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PRODUCT ENGINEERING

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Drawn Sections—Tubular Design
Spot Welding and the Design of
Built-up Sheet Steel Sections
Design of Cams for High Speed

•

Mechanical Design • Product Development

McGRAW-HILL PUBLISHING COMPANY, INC.

WHATEVER YOU NEED IN A MATERIAL ★

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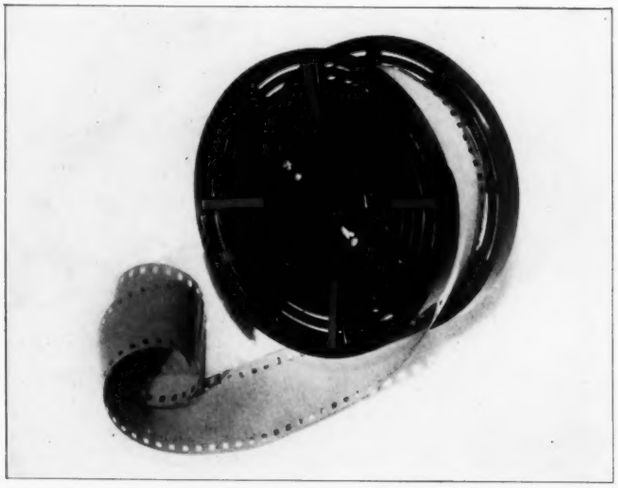

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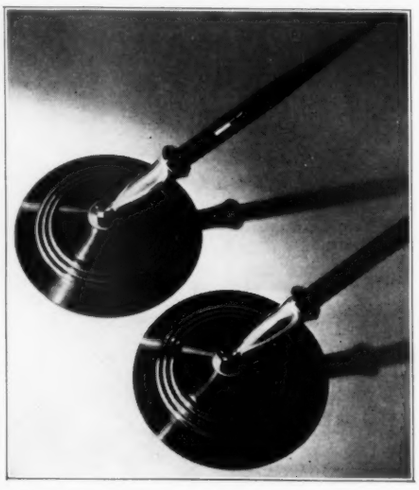

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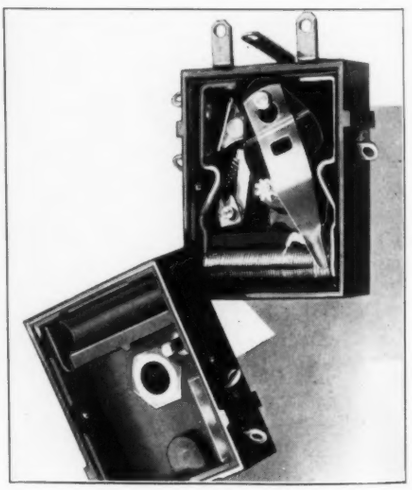

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The use of Durez for case and arm mounting on this precision resistor saved assembly costs and made possible improved design. Because of Durez' high dielectric strength, it made other insulation unnecessary.

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PRODUCT ENGINEERING

VOLUME 5 NUMBER 7

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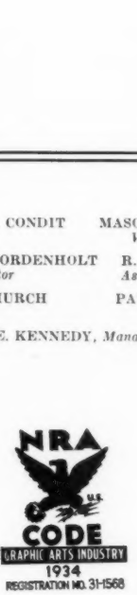

Coming Issues

In the August number we present the first of our articles on light-weight construction—the pros and cons of magnesium alloys as reflected by the experiences of engineers who have used the metal.

One of the vexing problems in the selection of electric motors is to figure WR^2. A short article will explain the why of this factor and how it is calculated in the simplest terms.

Other forthcoming articles will deal with motors and controls, impact extrusion of aluminum, design of gear trains and a presentation of the problem of grease versus oil lubrication for anti-friction bearings.

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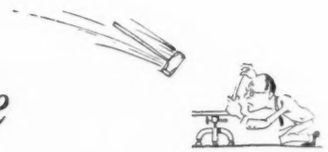
Intimate Correspondence



Invention Clearing House

PRODUCT ENGINEERING
NEW YORK

Several years ago I heard a statement that left a strong impression on me. It was to this effect: "We are now in the machine age, we are going into the mental age, and from the mental age we will go into the spiritual age."

Having an inventive turn of mind, and having made philosophic inventions over a period of years, I have concluded that a system should be established to hasten the coming of this spiritual age. With this system highly developed, ideas could be sent in to a central point for automatic research and adoption or rejection depending on whether they were an improvement over existing products or methods.

If this proposed system were a reality today I would turn in the following ideas for consideration: A milk plant without cows (this idea occurred to me about 20 years ago); Two meals a day, prepared in a central kitchen under sanitary conditions, with about as many styles and sizes as we have today, each meal to be packed individually and delivered to the consumer at meal time; Babies to be born in bottles and raised in central scientific institutions in order to improve the race (I think that no child can receive the best bringing up under present conditions, and that means must be worked out to associate the right adults and the right children); A highly efficient and foolproof system to mate folks to marry; Wearing apparel styles founded on moral principles; Scientifically designed houses built as automobiles are built and transported through the air to destination.

I realize that most of these ideas would look like some old one-lungers if they were put through the proposed system. But as it is well to know what we are doing in advance, lest we produce another Russia, some of these ideas may not be so bad to start with. *Product Engineering* and the system our President has for handling his mail look to me like forerunners of this system. It seems to me we are about as far along now with our development of it as were our present electrical devices 100 years ago. But our progress momentum is accelerating.

X. Y. Z.

WHO says invention is on the wane? The foregoing brief extract from a long letter is sufficient proof. We are withholding the name of the correspondent because some parts of the proposed system do not seem to be entirely worked out. His ideas, however, are certainly anything but one-lungers. They impress us more as multi-cylinder, automatically functioning mechanisms of a high order of development—babies in bottles, for instance.

As for *Product Engineering,* it is flattering to be looked on as a forerunner of the spiritual age, but we question if we are even ready for the mental age. Most of our thoughts and most of the things we print are still pretty closely tied up with machinery and the mechanical age. After all, there are some parts of *that* age that don't seem to be in such good control.

"His Master's Voice"

PRODUCT ENGINEERING
NEW YORK

Would you send me six copies of the reference sheet published in May, 1934 number? Mine is borrowed so often at the office, that the darn thing is going to be all worn out before it finds out who owns it. This may sound strange for the engineering department of a manufacturer of all-metal airplanes but the beam and torsion charts are excellent for a quick check before rigorous analysis is gone into.

Thank you.

HOWARD FIELD, JR.
Los Angeles

WE would, and did, with pleasure. But it looks, from this distance, as though that engineering department ought to have another subscription or so. The matter has been referred to our circulation manager.

Springs

PRODUCT ENGINEERING
NEW YORK

If possible, we would like to obtain two reprints of the article entitled "Charts for Obtaining Spring Dimensions Directly," as given in the June issue of *Product Engineering.* Please send these, together with any charges, to the attention of the writer.

E. D. LILJA
Barber-Colman Company

AS stated many times on this page there is no charge for clippings of pages wanted for reference files. It is only fair to observe, however, that the supply is limited and that we have to resort to photostats, for which there is a small charge, after the clippings are used up.

Page 2 Reader

PRODUCT ENGINEERING
NEW YORK

We would like to take advantage of the offer made on page 2 of *Product Engineering* for June, 1934, in which you say that you are in position to supply clippings of articles appearing in this magazine without charge. We would like to have "Charts for Obtaining Spring Dimensions Directly," which begins on page 215 of the June issue.

W. L. CHANDLER
The Fulton Sylphon Company

Another One

PRODUCT ENGINEERING
NEW YORK

We would like to secure clippings of the articles entitled "Charts for Obtaining Spring Dimensions Directly," and also "Determining the Critical Speeds of Shafting." The two above mentioned articles appear in the June issue of *Product Engineering,* pages 215 and 221.

According to the articles contained on page 2 of the same issue, the clippings of these articles will be furnished gratis upon request. We would therefore appreciate your forwarding these clippings, marked for the writer's attention.

W. H. RYAN
Kimberly-Clark Corporation

MESSRS. Chandler and Ryan were among the early inquirers for clippings of the spring articles so they got them. But the warning above should be heeded. The supply really is limited.

Whole Issue

PRODUCT ENGINEERING
NEW YORK

We have just read with a great deal of interest the article appearing in the June, 1934 issue of *Product Engineering* by C. H. Leis, Chief Engineer, Johnson Bronze Company, entitled "Bronze Bearing Alloys—Their Properties and Applications."

Will you be good enough to send us a copy of this issue for our files?

Thanking you.

O. L. DOWNS
The Superheater Company

WE can even take care of a few requests for whole issues. But here, too, the supply is limited because our list of paid subscribers is growing so fast that it is hard to keep enough copies on hand to supply new ones.

PRODUCT ENGINEERING

KENNETH H. CONDIT, *Editor*

MCGRAW-HILL PUBLISHING CO., INC. JULY 1934 VOLUME FIVE, NUMBER SEVEN

How Do You Make Your Inquiry?

ENGINEERS of motor manufacturers who are called upon to recommend motors for their prospects and customers seem to be in unanimous agreement that it is almost impossible to get mechanical designers to present their motor inquiries intelligently. It seems that although the product engineer will insist upon all the facts relative to the mechanical problem before he makes his decision, he expects the electrical engineer to sense motor requirements with only meager or even unreliable information about the application.

Interviews and correspondence with about fifty motor manufacturers compel the conclusion that they are quite justified in their complaint about the unsatisfactory manner in which product engineers make their motor inquiries. Indeed, it was the result of such an investigation that impelled us to devise the Motor Inquiry Form that appeared in our April number and was reprinted for free distribution among our readers.

Going further into the problem we investigated the manner in which engineers made their inquiries for motor controls, chain drives, gears and similar parts. Almost invariably we found the same situation. Information pertaining to operating conditions, speeds, direction of rotation, maximum loads and similar facts is usually lacking or incomplete when the inquiry is made.

It is hard to understand why so many engineers who pride themselves on technical accuracy, and boast that all their conclusions are based on facts, will transact business in a slip-shod manner. There are those who will demand every scrap of available information before they proceed with the design of their product. And then they make inquiries for "a quarter horsepower 1,800-r.p.m. motor," and let it go at that.

The ironical feature of this whole situation is that any failure of a product because the wrong part or material was used does not reflect on the manufacturer of the part or material, but is a black mark against the engineer who designed the product. For his own protection, the product engineer should at all times give full detailed information when he makes his inquiries.

Apprenticeship Needed

RECOGNITION of the importance of machine design in college courses is rapidly increasing. A movement instigated by Prof. Frank L. Eidmann of Columbia has resulted in the formation of a Machine Design Division in the Society for the Promotion of Engineering Education. At the convention of this Society held at Cornell during the week of June 18-23, a meeting of the Machine Design Division was attended by more than fifty professors representing colleges scattered all over the country.

A paper by Prof. E. S. Ault of Case School of Applied Science ably presented the question of advanced courses. In both the formal and informal discussion following this paper it became evident that the professors were concerned more about their present undergraduate courses rather than contemplated advanced courses. Product engineering is going ahead so rapidly that it is difficult to keep courses abreast of the times. This is further complicated by the limited time available for teaching.

It has been suggested to adopt the European system of requiring several years apprenticeship in engineering before the student is allowed to enter college or university. Such a requirement would be a long step in the right direction. It would make the student better prepared to master, and would give the colleges more time to teach, the fundamentals of engineering. It is with respect to fundamentals that the American graduate is sadly lacking when compared with the Europeans.

Light-Weight Design With Shot-Welded Stainless Steel

Explaining the shot-welding of stainless steel sheets and the principles used in the design of members and structures made of formed sheets joined by welding

COLONEL E. J. W. Ragsdale, chief engineer of "Hi-Tensile Division" of the Edward G. Budd Manufacturing Company, is credited with the statement, "Weight reduction can usually be procured in the same design merely by the substitution of a lighter material. But light-weight design signifies a more efficient disposition of a better material with full advantage being taken of the physical properties of that material and the methods available for its fabrication and assembly." And as pointed out by Albert Dean, structural car designer of the same company, "Airplane structures of approximately the same size, weight and stiffness can be designed to be made of spruce, aluminum or alloy steel. For each material certain design principles and methods of construction must be adopted."

For given load conditions and cross-sectional area of a structural member, its strength and stiffness will be determined by the disposition of the material, the yield point of the material, its modulus of elasticity and the method used to join the parts of the member and connect the member to other members of the structure. In some members, such as beams, stiffness will be the dominating requirement, which may result in the strength being far greater than necessary. Members that will always be in tension are designed solely for their area, while those that may be subjected to both

Based on interviews with engineers of the Edward G. Budd Manufacturing Company. The second and concluding installment of this article will be published in the August number.

Fig. 1 — All structural members in this car frame, the rear section of the Burlington "Zephyr," are made of formed stainless steel sheets joined by shot-welding. Below the window openings can be seen the aluminum foil used for heat insulation

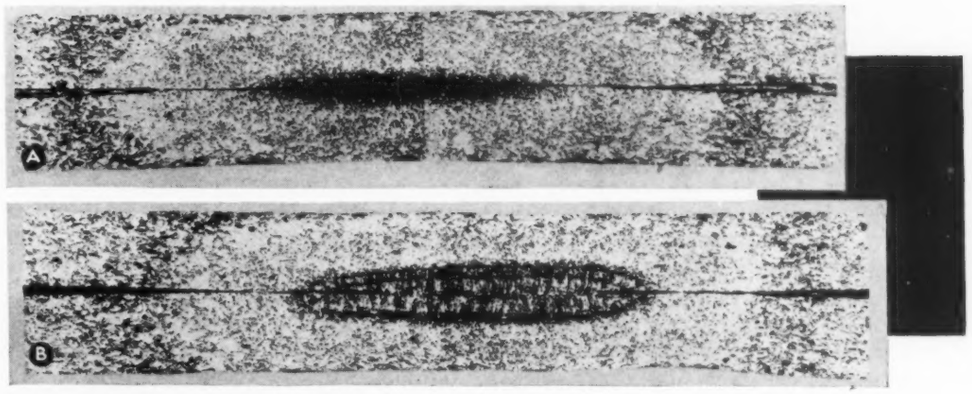
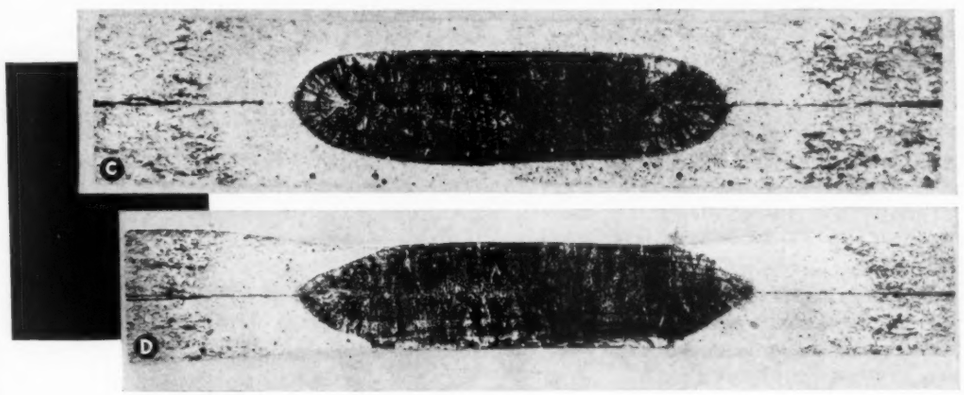


Fig. 2 — Photomicrographs (40×) of the cross-section through spot welds, showing how fusion progresses from the contact surface between the sheets to the outer surfaces. At A is shown the beginning of the fusion.....



.....At B the fusion has progressed into the sheet, but not far enough to make a good weld. C shows a normal weld and D shows too much fusion

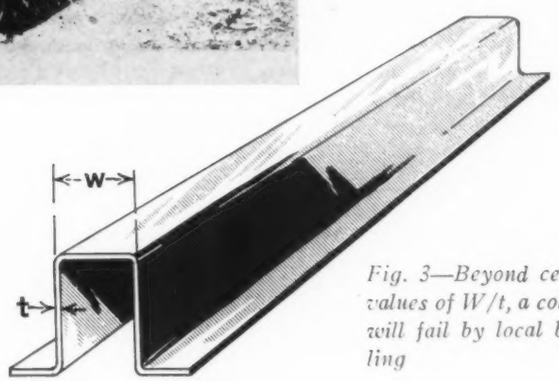


Fig. 3—Beyond certain values of W/t, a column will fail by local buckling

tension and compression by reversals of loading must be designed for both area and column stability.

With reference to the methods used to build up and join members for light-weight design, it is generally recognized that there are many advantages in using welding. It makes possible a better distribution of stresses and with welded joints it is easier to avoid eccentric loading. Also, welding does not involve the addition of useless material, such as rivet heads.

Stainless steel, 18 Cr-8 Ni, makes an excellent material for light-weight construction. It does not respond to a hardening heat-treatment, hence welds of this material are ductile and tough. The strength and hardness of the material are increased by cold working, an ultimate strength of 150,000 lb. per sq.in. and a yield point of 120,000 lb. per sq.in., with an elongation of 12 per cent being readily attainable. And, because the material is corrosion resistant, additional thickness need not be provided to take care of rusting.

Up until about four years ago the use of 18-8 stainless steel for light-weight design was greatly hindered because there was no available method of welding the material without impairing or destroying its corrosion and fatigue resisting properties. It was not until the development of the shot-welding process by engineers of the Edward G. Budd Manufacturing Company that there was available a method of commercially welding 18-8 stainless steel without in any way impairing its strength or corrosion-resisting properties.

Loss of corrosion resistance in stainless steel welds, as explained by Dr. Leiter of the Budd Company, is brought about by the precipitation of chromium carbide at the grain boundaries when the metal is exposed to a critical temperature. This precipitation begins at a slow rate when the metal has reached some 900 deg. F. and progresses at a faster rate as the temperature increases to about 1,500 deg. F. One would infer that because of the low carbon content of the steel, only a small amount of carbide would be precipitated. But the carbon combines with about eight times its weight of chromium, this accounting for the relatively large amount of chromium carbide that will be precipitated. And by this precipitation, the metal adjacent to the grain boundaries will be robbed of its chromium. As it is the chromium that makes the metal corrosion resisting, the metal at the grain boundaries if impoverished of its chromium will be readily attacked by acids and other corrosive media. Corrosion will progress along the grain boundaries until nothing remains of the metal except a crumbly mass of loosely held together grains.

After much painstaking experimenting and study, engineers of the Budd Company arrived at a solution of this problem that is amazing in both its simplicity and effectiveness. The first step was a study of the

speed of precipitation of the chromium carbide. It was found that this was sufficiently rapid to cause trouble in the then existing types of welding.

In the photomicrographs in Fig. 2, is clearly shown how the spot-welded "ingot" develops, the fusion temperature first being reached in the plane about midway between the points of the electrodes. Fusion temperature is reached at this mid-plane first, whether the electrodes are gripping a single sheet or two overlapping sheets. But in the latter instance, the somewhat higher electrical resistance at the boundary surface between the lapped sheets develops heat at a faster rate, causing the "ingot" to mushroom as shown in the photomicrographs.

The reason for the "ingot" starting in the mid-plane between the two electrodes is explained by the rapid heat losses at the areas contacted by the electrodes. This keeps the metal at the surfaces cooled, thereby establishing a gradient of increasing temperature toward the middle of the sheets, or the point midway between the two electrodes. Several factors will determine the steepness of the gradient, the main factors being the size of the spot-welding electrodes, the magnitude of the spot-welding current and the electrical resistance in the path of the welding current. One favorable factor for the establishment of a high temperature gradient in the welding of stainless steel is its relatively high electrical resistance. This causes heat to be liberated in the metal at a much faster rate than would be possible with the same current density in plain carbon steels, allowing a correspondingly shorter welding time.

From the foregoing it is evident that if the metal forming the spot-weld "ingot" is brought to the fusion temperature with great rapidity and then cooled rapidly, there will not be time for the precipitation of chromium carbide. This is exactly what is accomplished in the Budd "shot-welding" process. By the automatic control of the duration of a predetermined intensity of spot-welding current, the weld "ingot" is formed and cooled in a minimum time and the zone of fusion temperature is controlled so that the "ingot" penetrates to a depth of from 50 per cent to 80 per cent, as may be desired, of the thickness of the sheet. This process requires a high welding current for a definitely timed short interval.

Properties of Stainless Steel Shot Welds

When the weld has been completed by the shot-welding process, the metal forming the weld "ingot" has been subjected for such a short time to a temperature in excess of 2,500 deg. F., that there has been no time for an appreciable precipitation of chromium carbide. The metal at a distance of one sheet thickness or more beyond the "ingot" never reaches the temperature at which chromium carbide begins precipitating. Therefore the narrow band surrounding the ingot has been exposed to the critical range for such a short time as to be practically unaffected and the finished weld will be as corrosion resistant as the parent metal. This has been substantiated by innumerable tests in the laboratories of the Budd Company and elsewhere.

Concerning the properties of a properly made shot weld, because the 18 Cr-8 Ni steel remains austenitic from the molten state down to room temperature, the shot-weld "ingot" will be as ductile as the parent metal in the annealed state. This is strikingly illustrated by joining two strips with a single shot-weld. The strips can then be twisted 90 deg. to 180 deg., depending on weld size, with respect to one another without breaking the weld. The shear strength of the weld will be about 90,000 lb. per sq. in.

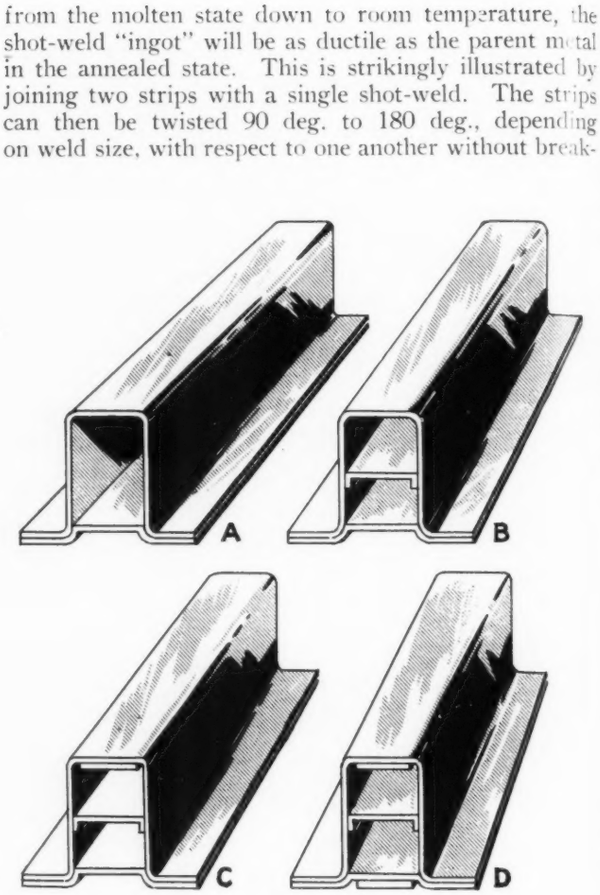


Fig. 4—Post sections using a flanged channel for the basic section. The addition of shallow channels and flat strips as shown at B, C and D successively increases the strength

Development of the shot-welding process for stainless steel has opened an entirely new field of possibilities in the use of this material for structural design. Structural members can be fabricated from relatively thin sheets which are first bent or drawn to the desired shape and then shot-welded together. And because the designer is not restricted to the use of standard shapes, as when using hot-rolled sections, he is free to design the member to whatever shape will give him the required strength and stiffness with minimum material.

A column or compression member formed out of a flat sheet, as shown in Fig. 3, can be considered as being composed of a number of flat sheets supporting one another along longitudinal edges. The resistance of such flat sheets to buckling varies inversely as the width of the sheet and directly as its thickness, the ratio of the unsupported width of the flat sheet to its thickness being designated as the flat pitch. As the flat pitch increases, the buckling stress will be lowered.

In the design of all columns, in order to minimize the column effect the largest possible radius of gyration must be employed. In the types of columns de-

scribed here this can be obtained by bending a relatively thin sheet to form a large section. But care must be taken to avoid flat surfaces having a flat pitch so large that local failure will occur through buckling of a flat surface at a load considerably less than the theoretical column strength. For every design of column there will be a maximum flat pitch beyond which the column will fail by "crinkling" of the flat surface rather than failure of the column as a unit.

Failure by crinkling of a "flat" is caused by elastic deformation. All elastic deformation is inversely proportional to the product of the effective moment of inertia and the modulus of elasticity (the effective *EI* value) of the part affected. Therefore, to increase the strength of a given flat against crinkling, the moment of inertia of the flat must be increased. This can be done by welding on an extra flat piece, as for local reinforcement, or welding an angle section to the flat element. Both the area and the moment of inertia will be increased thereby, thus not only increasing the effective *EI* value, but also decreasing the unit stress.

Another method of increasing the strength of a column against failure by the crinkling of a flat surface is to break up the flat surface by means of grooves. By breaking up the flat surfaces in this manner, the flat pitch is decreased without increasing the thickness of the metal at any point.

One important precaution is that where a stiffening angle or groove is used, the elements of the composite flat width must have a flat pitch in themselves not higher than the equivalent flat pitch of the composite element. Thus, the flat pitch of the outstanding leg of an angle can be considered with sufficient accuracy as twice the flange width divided by its thickness.

In the design of shot-welded members and structures certain general rules and precautions which follow riveting practice must be observed. Welds should be located in the middle of the flanges, and more than two planes should not be welded with a single piece. Welds in tension or under a peeling action should not be used and thin flat sheets that might drum or feel flimsy should be avoided. If a thicker sheet is not desired, the thin sheet can be corrugated or reinforced.

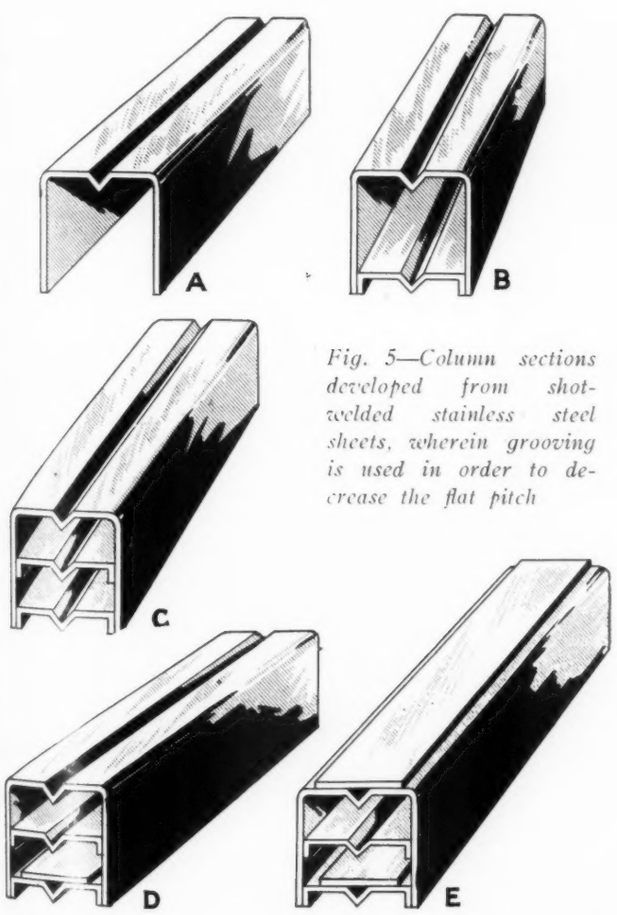


Fig. 5—Column sections developed from shot-welded stainless steel sheets, wherein grooving is used in order to decrease the flat pitch

Typical Built-up Columns

Perfect nesting fits cannot be obtained from parts bent on a brake, therefore the design should avoid such requirements unless drawn sections are used. Where carbon steel or aluminum will come in contact with the stainless steel, a protective insulation of paint should be specified. This also applies to plated parts as well because stainless steel is electropositive to most other materials and will corrode the others by electrolysis.

With reference to the drawing room practice, the minimum number and strength of welds should be noted for all joints and should also be indicated on the assembly drawing.

Applying the principles of design relative to flat pitch surfaces and after determining the desired strength and spacing of shot welds, various shapes of drawn sections can be designed for posts or columns, or sections can be combined to give columns of the required strength or stiffness. Using stainless steels, extra thickness is not required to provide for rusting. And because the flat sheet can be bent into any desired shape, the designer is not restricted as in the case of rolled steel, to the use of standard sections.

A simple type of post or column section is shown in Fig. 4. The addition of a shallow channel to the basic section materially increases the strength of this column against local loads. The embossing of the cover plate not only increases its stiffness, but also serves as a gage in the fabrication. At *C* and *D* in Fig. 4 are shown two further developments.

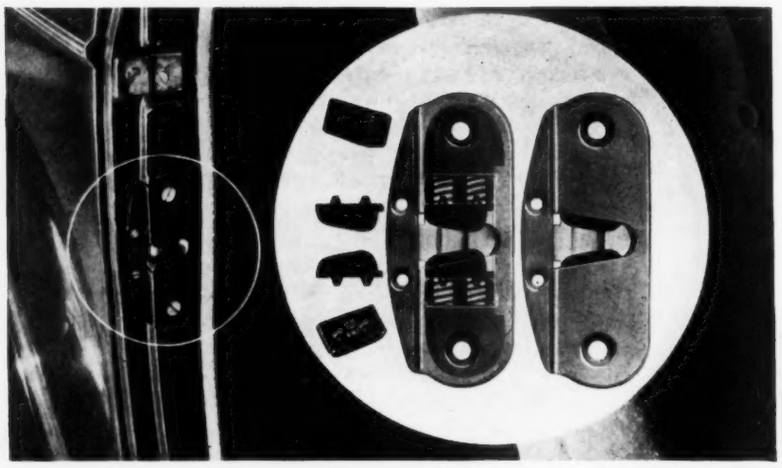
In Fig. 5 is illustrated a series of column sections built up with modified channel sections and flat reinforcing plates. At *A* is shown the channel section with a V-groove along the center of the flat, the purpose of this groove being to reduce the flat pitch of the element. The addition of the grooved channel section as shown at *B* supports the flanges of the basic section, and increases the cross-sectional area. In the composite section shown at *C* the area has been increased further and the flat pitch of the flanges of the basic construction has been cut in half, resulting in a greater proportional increase in the strength of this section when used for short columns. At *D* and *E* are shown how the addition of flat plates further increases the strength of the column. When this section is used as a column it is given lateral support at short intervals where possible, thereby cutting down the column length and thus permitting higher working stresses.

PRODUCT DEVELOPMENTS

• *Plastic Molded Door Bumpers*

After tests in laboratory hot rooms and cold rooms to determine any effect of heat and cold on assemblies, and after extensive road tests conducted on automobile proving grounds, molded bumper shoes in the door dove-tail assemblies have been adopted for several makes of cars.

This development grew out of the search by automobile engineers for a door bumper casing assembly that would be free from squeaks and rattles for the life of the car, without oiling or greasing. Previously, the entire assembly had been made of metal, which wore relatively rapidly and permitted excessive play that caused rattling.



Plastic molded door bumper design now being applied to automobile bodies. To reduce friction, a special molding compound impregnated with graphite is used

It is a peculiarity of synthetic plastics that when in contact with wearing surfaces of metal, as in gear trains, there is a marked reduction of wear on both materials. Preliminary tests duplicating the impact and friction of door bumper shoes against the wedge plates showed a similar reduction in wear.

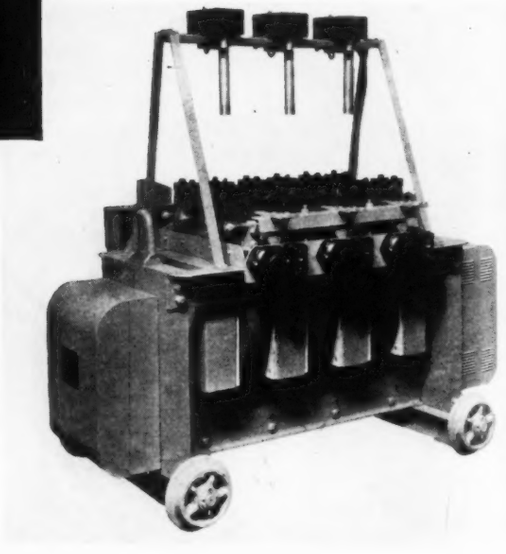
In developing the first molded shoes, the fabricator, Chicago Molded Products Corporation, found that graphite could be incorporated into the molding powder to further reduce the coefficient of friction and ease the entry of the wedges. A special type of extra-strength Durez molding compound was decided upon after tests.

• *Photocells for Temperature Control*

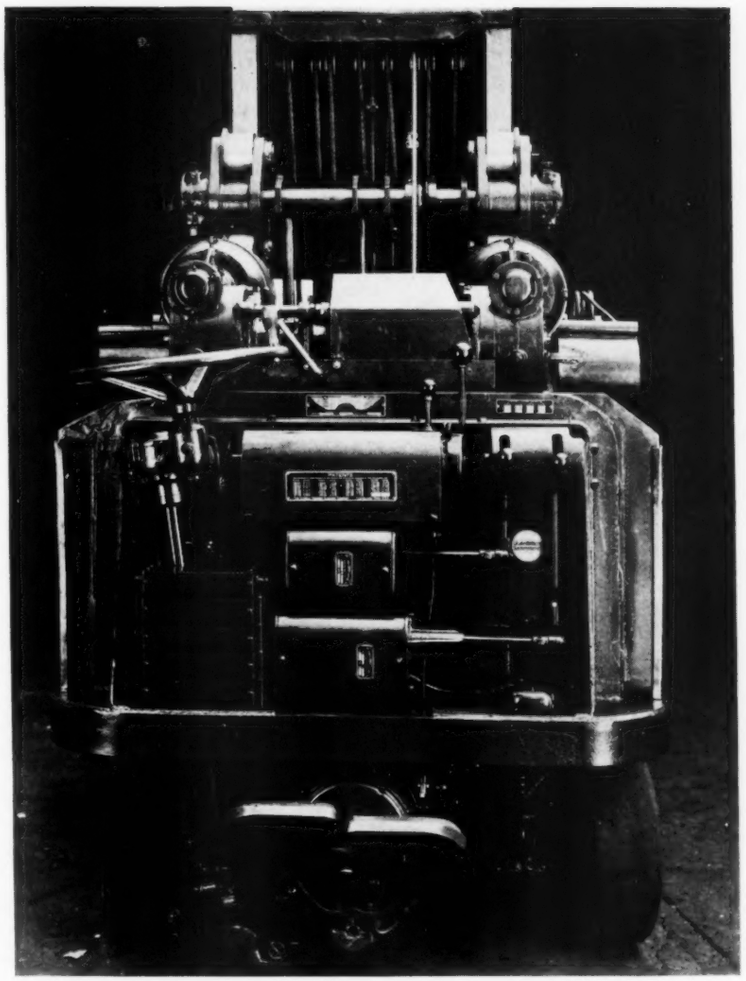
In the latest designs of the Berwick electric metal heaters manufactured by the American Car & Foundry Company, four of which have been installed in one of the leading automobile plants for making drag links, photocells are used to automatically control the temperature to which the metal is heated.

As can be seen in the accompanying illustration of a three-electrode heater, a photocell is mounted above each of the heating units so that the light from the metal being heated will be projected upon the light-sensitive cell. In operating the heater a piece of steel is put in the machine and heated to exactly the desired temperature as determined by an optical pyrometer or other suitable means. The photo-electric control unit is then adjusted to control the current supply to the heating electrodes so that the metal will be heated to that temperature. According to tests, the photo-electric unit automatically controls the temperature within a variation of a few degrees.

Because the operator is entirely relieved of the necessity of paying attention to the temperature to which the stock is being heated, and because of the accuracy of the photocell control, the stock is always heated to the same temperature. Breakage of forging dies because of underheated stock is avoided, and there are no losses



Photocells are used to automatically control temperatures in the Berwick electric metal heater



The latest Elwell-Parker electric truck has one motor to drive the truck, a second one to operate the platform hoist and a third to push or pull the load off or on the truck platform

from overheated or burned stock.

Another advantage of the electric heating is that less scale is formed, resulting in longer die life and smoother forgings.

• *Electric Truck Has Compensating Wheels*

Two noteworthy features characterize the latest electric truck designed and built by The Elwell-Parker Electric Company for a Detroit automobile body plant to handle 25-ton body dies. One of these features is the complete electrification and control of the truck, and the other is the design of the rear wheel mountings.

In addition to the motor for driving the truck, there is another one for raising and lowering the platform and a third one for the die-pulling unit for loading and unloading the truck.

The motor for raising and lowering the load platform drives a hoist drum through a worm and gear and heat-treated spur gears. From the hoist drum a ½ in. plow steel cable runs over sheaves at the top of the elevator uprights and then down to the elevator carriage assembled integral with the load platform. Safety limit switches prevent over travel of the platform and electromechanical brakes automatically hold the load in any position when the power to the motor is shut off.

To compensate for unevenness of the floor over which the truck is operating, an unusual design of rear axle mounting was developed. All four trail wheels must steer simultaneously and are mounted on a one-piece axle casting. A horizontal rocker shaft runs through the center of the axle casting, perpendicular to the axles. The rocker shaft in turn is mounted in a cast steel trunnion and is free to rotate about an axis parallel to the wheel axles. Thus the axle casting is free to swivel about a horizontal axis parallel to the wheel axles and also a horizontal axis vertical to the two wheel axles.

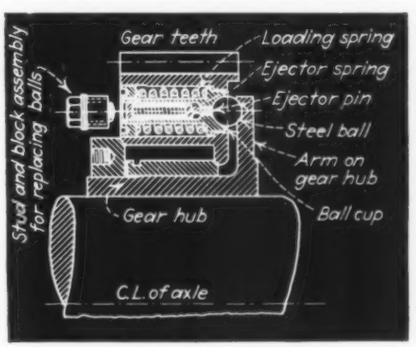
With the axle construction described, the action of the wheels in going over rough places is somewhat analogous to the action of the wheels on the equalizing trucks used on railroad cars. With this type of construction the amount of vertical rise of the truck body resulting from a wheel going over a bump is only half as much as would occur with a plain swivel truck.

To reduce the unit floor loading, each of the four trail wheels are equipped with two 18 x 12 in. solid rubber tires.

• *Safety Gear for Axle Generators*

To protect against the contingencies of a "jammed" generator axle or "jammed" gear and pinion, the General Electric axle-mounted generators are provided with a "safety" gear. This gear is designed as shown in the illustration so that in the event of the generator armature locking or being excessively overloaded, or if the gears are jammed, the axle is freed from the gear and continue to rotate.

In its normal operating position, as shown in the accompanying fig-



Design of safety gear to automatically free the gear from its axle in case of overload

ure, there are six ball and spring assemblies located in as many pockets in the rim assembly and spaced equi-distantly around the center. Under normal operating conditions the balls are held against and in the sockets that are machined in the arms of the hub. The characteristics of the spring and the design of the sockets and balls are such that the unit will not shear because of any coupling shock load that the car structure can withstand, but will separate if the slipping point of the wheels is approached. This last condition is such as would occur with a locked armature.

In the event of a mechanical failure in the generator or its gearing, tending to prevent rotation of the car wheels, the shearing effect of the hub arms on the balls forces the balls in against the springs, thereby permitting the hub to turn free of the gear rim. The auxiliary spring insures removal of the balls from the cups after the balls have been forced out of the hub sockets by excessive torque.

The seats on the hub member on which the gear rim rests in normal service are so designed that when the balls have been dislocated, the gear rim moves to the ring and downward, with the gear face resting on the gear case and clear of the rotating hub. The gear is, thereby, inherently "trip-free" in its operation and hence no renewal of parts is necessary after running for long periods with the balls displaced. Adjustment of the "pull-out" value of the gear can be obtained, if desired, by varying the position of the plugs in the gear rim sockets, which is simple to do.

• *Fireless Locomotive Has Welded Insulated Tank*

In the "fireless locomotives" shown here, designed and built by the Heisler Locomotive Works, the "boiler," or tank as it is more properly called, is filled about ¾ full of water. Steam at about 100 lb. per sq.in. pressure, or higher, is then injected below the water line, heating the water to the boiling temperature and raising the pressure to equal that of the injection steam. The heat energy thus stored in the water and in the material of the tank is then used to drive the locomotive. The locomotive has large cylinders and small driving wheels so that relatively low steam pressure in the cylinder will give the required tractive effort.

Steam is admitted to the cylinders through a conventional throttling valve. As the steam pressure in the tank drops the boiling temperature of the water drops correspondingly and steam is generated, using the heat stored in the water. Thus boiling continues until the decreasing water temperature and increasing steam pressure and temperature are again in equilibrium.

In order to minimize radiation losses, the storage tank of this locomotive is insulated with magnesia blocks 2 in. thick, wired in place. The ends of the tank are protected with loose rock wool, the whole tank being encased in a sheet-iron jacket. Test data show the pressure drop because of heat radiation losses is approximately one lb. per sq.in. per hour.

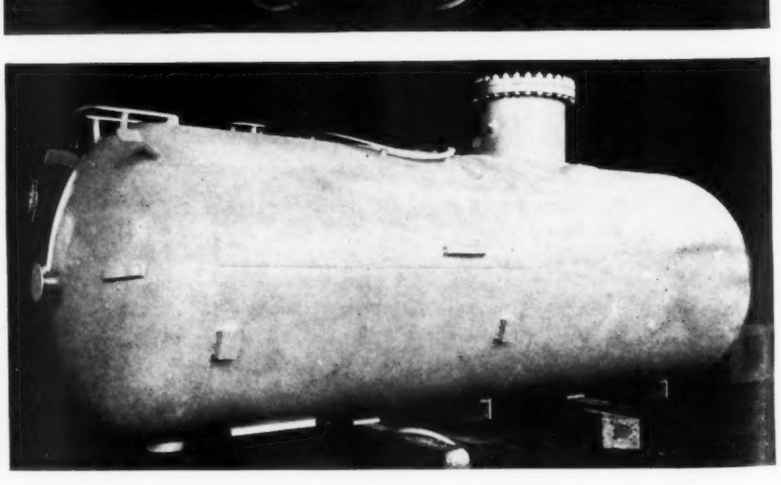
One of the problems in this design was to design a boiler that would not develop leaks. Because of the insulation and the steel jacket leaks would not become evident until after much damage had been done to the insulation.

Because of importance of guarding against possible leakage it was decided to abandon the conventional riveted tank and developed a welded one. The all-welded tank pictured here was designed. It is 69 in. in diameter and is safe for a working pressure of 215 lb. per sq.in. It was built by Struthers-Wells Company, using the Lincoln shielded arc process.

In order to further increase the efficiency of the locomotive the driving axles are mounted in Timken tapered roller bearings.

• *V-Belt Drives for Veneer Driers*

In their new design of veneer driers, the Merritt Engineering & Sales Company, Inc., use a V-belt drive, the same type of drive being recommended for their motor-



Fireless locomotive which gets its steam from heated water carried in its welded tank. Small driving wheels and large cylinders give large tractive effort at low steam pressure

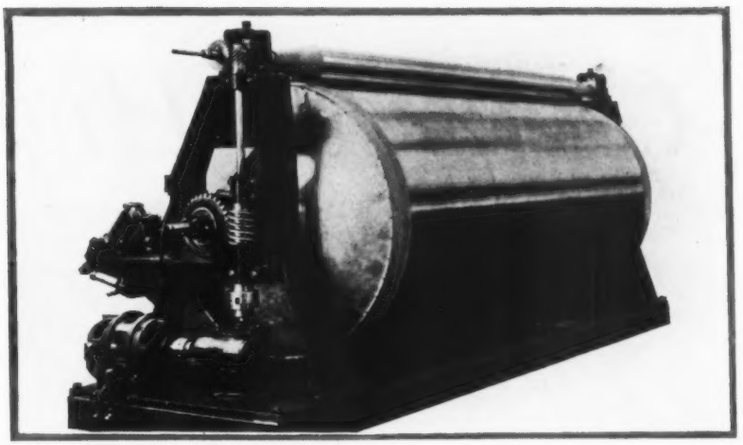
driven veneer lathes. Quoting Richard W. Stanley of the engineering department of Merritt Engineering & Sales Company:

"For the main spindle drive from motor to drive shaft, silent chains have proved quite satisfactory. With the advent and broadening application of multiple V-belt drives, certain advantages in their use have been demonstrated.

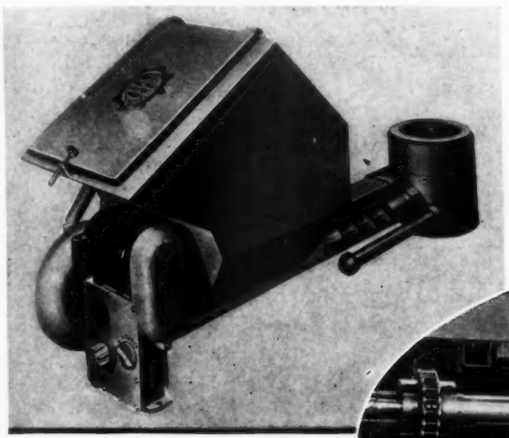
"Probably the most outstanding advantage of the multiple V-drive, over the silent-chain drive, lies in its more natural adaptability to the service imposed by the multi-speed, constant-horsepower type of motor. With this motor it is required to design the drive within reasonable limits of pitch-line velocity when the motor operates at its highest speed and, conversely, the belts or chain must be designed for sufficient capacity to transmit the full motor overload rating with the motor running at its lowest speed, which is usually one-third of the top speed. In a silent-chain drive for veneer lathes and driers these requirements at once limit the top motor speed to about 1,200 r.p.m. and make the use of an oil-tight housing imperative.

"A multiple V-belt drive, on the other hand, can still be designed for sufficient capacity at the low speed while not exceeding the limits of allowable belt velocity at the high speed. In addition, the use of motors having a top speed of 1,800 r.p.m. instead of 1,200 r.p.m. can now be used, permitting a substantial saving in the initial motor cost. Furthermore, the usual reduction ratios, lying between 3.75 and 5.0 to 1, permit the use of a V-flat drive, the grooveless driven sheave further reducing the initial cost of the machine. Inherent cushioning effect and reduced maintenance attention add still more to the desirability of the V-belt drive.

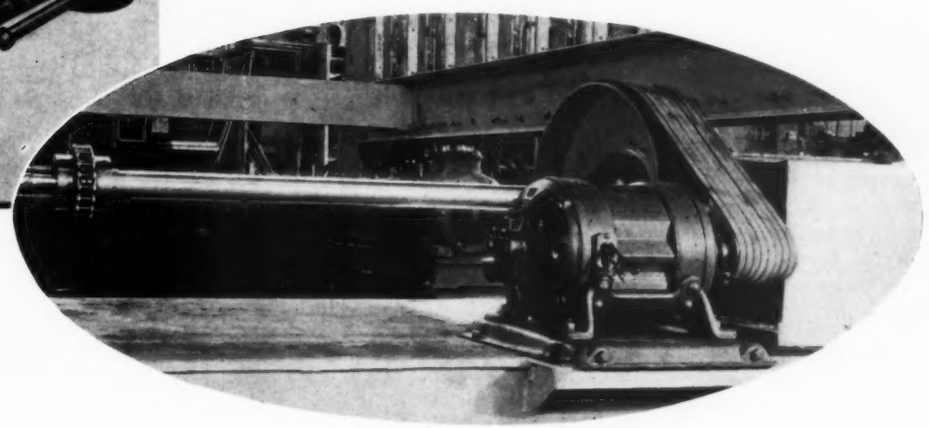
"Whether, of course, a veneer lathe drive takes the form of a silent-chain drive, multiple V-belt drive, or a gear-reduction unit, the most important factor in their design is a thorough knowledge of the service to be encountered, which, in the case of veneer lathe operation, can often be extremely severe.



Arc welded rotary drum dryer built by Struthers Wells Company with stainless steel main shell and high-carbon steel heads insulated and lagged with polished aluminum sheets



Five-speed transmission that can be shifted while in operation, fan silencer and positive lubrication are features of the Link Belt Company's new domestic stoker



V-flat drive used on the new design of veneer driers developed by the Merritt Engineering and Sales Company

• *Ignition Coils for Operating in Water*

In the latest design of ignition coils announced recently by Mallory Electric Corporation, a special arc-resisting Durez is used for the molded covers and the entire case is also molded of Durez to resist moisture and prevent high engine heats from damaging the interior windings.

It is claimed that this new ignition coil is the only coil ever made that will run with even the secondary terminal and entire case submerged in water. It is designed for motor-car, truck and marine use, and is said to be constructed so as to cushion the windings against shock. A special arc-resisting compound was used.

Cams for High Speeds–I

How to develop acceleration curves that will give a movement free from shock and how to determine the form of the cam to produce such acceleration

C. N. NEKLUTIN
Chief Engineer, The Universal Match Corporation

THE productive capacity of an automatic machine is limited in many instances by the inertia forces developed therein, which are a function of the masses and accelerations of the parts that have intermittent or reciprocating motion. And, because of the tendency to increase the speed of automatic machinery, a thorough knowledge of the methods of calculating inertia forces has now become more important than ever before.

Because the force of inertia is a product of the mass of the moving part and its acceleration, it can be decreased by the use of light strong alloys. But that fact does not make the selection of cam profiles of less importance. With the same materials and using the most suitable movements, a designer can still develop a machine of higher capacity, a factor which may be more desirable from the manufacturing point of view.

In designing a cam for a certain displacement the attention of the designer should be concentrated, first of all, on the acceleration and deceleration created by the cam, these being the source of the inertia forces. For high-speed machines the designer must therefore select a movement that has the smallest possible acceleration and is free from shocks caused by a sudden change in acceleration. Instead of calculating the acceleration of a cam of a certain shape, it is more proper to select the most suitable form of acceleration curve and then determine the form of the cam producing such acceleration.

Undoubtedly the most commonly used acceleration and deceleration curve is the sine curve as illustrated in Fig. 1. Considering a stroke as being the movement of the cam follower from one position of rest to the next position of rest, the base OE of the acceleration curve will represent the time necessary for one stroke. In so far as the sine curve is concerned, this distance is equal to 2π. During the stroke, the acceleration goes through one complete cycle as represented by the sine curve from 0 deg. to 360 deg., or from 0 to 2π if the angle is measured in radians. This is entirely independent of the angle of rotation of the camshaft and it makes no difference what angle of cam rotation may be required to produce the stroke. In so far as the sine curve is concerned, the abscissa OE in Fig. 1 is equal to 2π. The ordinates of the sine curve are the values of the acceleration. For any fractional part x of the stroke of the cam follower, one stroke being represented by the abscissa OE, the value of the acceleration y will be:

$$y = A \sin 2\pi x \qquad (1)$$

In this equation A is the maximum acceleration. The speed of the cam follower at the point t in Fig 1 is equal to the area OtC under the curve. If α is the angle in degrees through which the camshaft turns during the stroke of the follower, and N is the revolutions per second of the camshaft, it can be shown that the velocity v will be expressed by the equation:

$$v = \frac{A\alpha}{2\pi\ 360\ N}(1 - \cos 2\pi x) \qquad (2)$$

The curve for velocity is shown in Fig. 2, the ordinate tD in this figure corresponding to the ordinate tC in

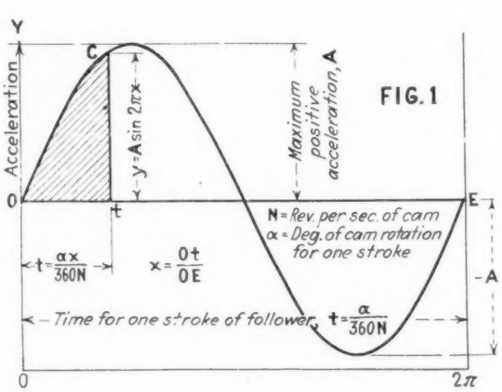


Fig. 1—Area OtC *under the sine curve of acceleration represents the speed of the cam follower at the point* t

Fig. 1. The variation in the radius of the cam, or the displacement of the cam follower corresponding to the point t is equal to the area OtD in Fig. 2. The equation for this displacement is:

$$S = \frac{A}{2\pi}\left(\frac{\alpha}{360\,N}\right)^2\left(x - \frac{\sin 2\pi x}{2\pi}\right) \qquad (3)$$

When $x = 1$ the displacement will be equal to the throw F of the cam, in inches, or:

$$S_{max} = F = \frac{A}{2\pi}\left(\frac{\alpha}{360\,N}\right)^2 \qquad (4)$$

Transposing this equation we have:

$$A = F\,2\pi\left(\frac{360\,N}{\alpha}\right)^2$$

If in the above expression S is expressed in inches, the maximum acceleration will be in in. per sec. per sec. By substituting this value of A in Equations (1), (2) and (3) we shall obtain the final expressions:

$$a = 2\pi F\left(\frac{360\,N}{\alpha}\right)^2 \sin 2\pi x$$

$$v = F\left(\frac{360\,N}{\alpha}\right) \sin 2\pi x$$

$$S = F\left(x - \frac{\sin 2\pi x}{2\pi}\right)$$

All these equations have the trigonometrical functions of angles measured in radians. But in handling these equations it is better to replace 2π by 360 deg., as tables in handbooks give the values for them measured in degrees.

Then the final forms of the equations will be:

$$a = 2\pi F\left(\frac{360\,N}{\alpha}\right)^2 \sin 360x \text{ deg.} \qquad (5)$$

$$v = F\frac{360\,N}{\alpha}(1 - \cos 360x \text{ deg.}) \qquad (6)$$

$$S = F\left(x - \frac{\sin 360x \text{ deg.}}{2\pi}\right) \qquad (7)$$

In these equations x is the fractional part of the stroke of the cam follower, F is the maximum throw of the cam in inches and S is the displacement of the cam follower in inches.

Where the line of movement of the cam follower passes through the center of rotation of the cam, the variation in the cam radius S will be equal to the movement of the follower.

From equation (5) the maximum value of the acceleration will be at the point where sin (360 x deg.) has its maximum value of unity. This will give a maximum acceleration:

$$A_{max} = F\,2\pi\left(\frac{360\,N}{\alpha}\right)^2 \qquad (8)$$

As shown by this equation, to decrease the acceleration the designer may decrease the stroke F or increase the working period α of the cam. The minimum value of F is determined by the necessity to perform a certain operation by the machine part moved by the cam, the limit for α being established by the necessity of performing other movements during the revolution of the camshaft.

Where possible it is better to have over-lapping movements, that is, to start the succeeding movement before the previous one has been completed fully, keeping in mind that the clearance between adjacent moving parts be as small as possible. This will result in an increase of α, which may be highly effective because the decrease of maximum acceleration is inversely proportional to the square of the increase in the working period of the cam.

In some instances the designer may increase the throw and make the clearance between moving parts larger at the beginning or at the end of the movements. By doing so the working angle α can be increased to such an extent that the acceleration will be smaller in spite of the increased F. This method will often allow decreasing the maximum acceleration or permit the speed N to be increased noticeably.

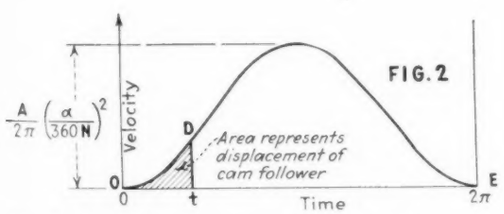


Fig. 2—Speed curve corresponding to the sine curve of acceleration. Displacement of cam follower is represented by area under the curve

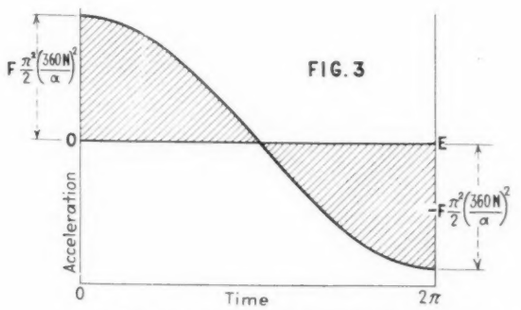


Fig. 3—Acceleration curve for the Scotch yoke mechanism, which produces the same movements as a crank with an infinitely long connecting rod

Scotch Yoke Movement

As is well known, in the usual connecting rod mechanism the motion of the crosshead is not harmonic, When the crank has made a quarter of a revolution from the dead center, the crosshead will have passed beyond its mid-position. This immediately shows that the velocities and accelerations are not the same for positions of the crosshead, equally distant to each side

of the mid-position. But as the ratio of length of connecting rod to crank radius increases, the motion of the crosshead becomes more nearly harmonic.

Pure harmonic motion is obtained in a crank mechanism when the ratio of length of rod to crank radius is equal to infinity. Under such a condition the motion of every point in the connecting rod is always parallel to the direction of travel of the crosshead. This relationship is attained in the familiar Scotch Yoke wherein the crankpin slides in a straight slot in a block or crosshead. The slot is perpendicular to the line of travel of the crosshead which slides between parallel guides.

Treating the Scotch Yoke as a cam, the acceleration curve as shown in Fig. 3 can be constructed and the equations of motion can be derived as for the sine curve.

The equations for a cam producing the movement such as created by a Scotch Yoke, are as follows:

$$\text{acceleration} = F\frac{\pi^2}{2}\left(\frac{360\,N}{\alpha}\right)^2 \cos 180x \text{ deg.} \qquad (9)$$

$$\text{speed} = F\frac{\pi}{2}\left(\frac{360\,N}{\alpha}\right) \sin 180x \text{ deg.} \qquad (10)$$

$$\text{throw} = F\frac{1 - \cos 180x \text{ deg.}}{2} \qquad (11)$$

Comparing the maximum acceleration of both these cams, the maximum acceleration equaling the maximum deceleration, for the Scotch Yoke:

$$A_s = \frac{F\pi^2}{2}\left(\frac{360\,N}{\alpha}\right)^2 \qquad (12)$$

For the "sine curve" cam

$$A_c = F\,2\pi\left(\frac{360\,N}{\alpha}\right)^2 \qquad (13)$$

Dividing one by the other we have:

$$\frac{A_c}{A_s} = 1.273 \qquad (14)$$

This shows that the maximum acceleration of the "sine curve" cam is 27.3 per cent higher than in the "Scotch Yoke" cam. But the latter has two points of shock, one at the start and one at the finish of the movement. These are caused by the abrupt change of acceleration. On the other hand, the sine curve cam does not have any points of shock.

Combination Scotch Yoke and Sine Curve Cam

Assume that the problem is to move a machine part carrying something from one station to another, but that no operation is to be performed on the product while it is in motion, but that immediately after coming to a stop there will be some other parts of the machine which will begin to operate on the part delivered to this station.

Suppose in this instance it is permissible to allow a shock at the start and to have a high rate of acceleration. Also assume it is advisable to have the smallest possible deceleration and a shockless stop, so that the product when delivered to the station will be disturbed as little as possible by inertia forces. For such an application a combination cam can be developed, using the crank movement during the acceleration period and the sine curve during the deceleration.

The total area under the acceleration curve between zero and any given abscissa is equal to the velocity at that point. Area below the X axis is considered negative, and the total area between the acceleration curve and the X axis plus that between the deceleration curve and the X axis must be equal to zero when added algebraically. Thus the positive area of acceleration must equal the negative area of deceleration.

Total positive velocity under the acceleration curve will be the integral of the acceleration curve between the limits of zero and x. Total negative velocity will be the integral of the deceleration curve between the limits x and 2π. Equating the two equations thus obtained gives the value of x that will give the proper division of the cam between acceleration and deceleration.

To fulfill this requirement the acceleration should exist during 11/25 of the working period and the deceleration during 14/25 of the working period of the crank movement and sine curve movement respectively. Under this condition the acceleration will be:

$$a = F\frac{\pi^2}{2}\left(\frac{360\,N}{\frac{22\alpha}{25}}\right)^2 \cos 180x \text{ deg.} \qquad (15)$$

$$= 6.372F\left(\frac{360\,N}{\alpha}\right)^2 \cos 180x \text{ deg.} \qquad (16)$$

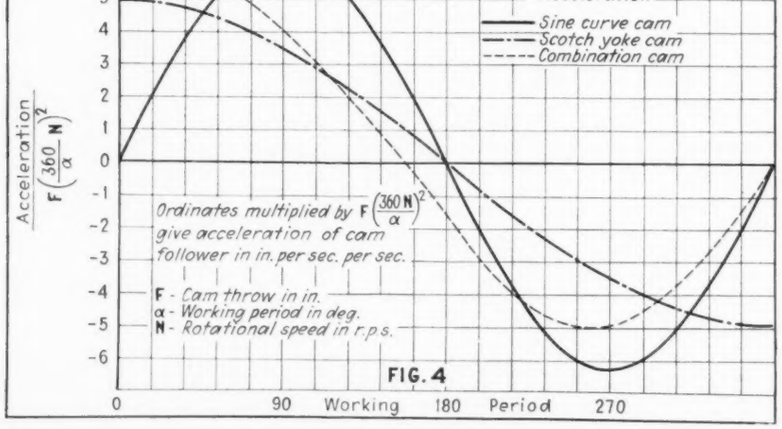


FIG. 4

Fig. 4—Superimposed curves of acceleration as produced by the sine curve cam, the Scotch yoke mechanism and the combination cam

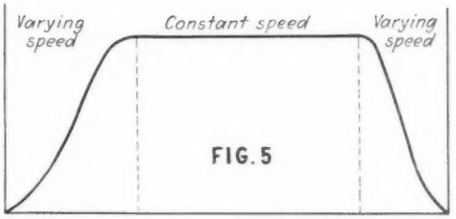
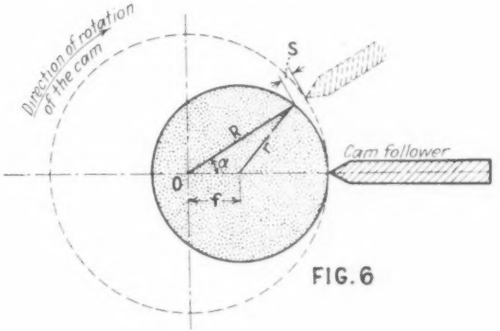


Fig. 5—Velocity curve for a cam to give a constant maximum speed during a portion of the stroke

Fig. 6—Eccentric disk with direction of travel of the follower passing through the axis of rotation of the disk



Similarly, the deceleration rate will be given by:

$$d = F\,2\pi\left(\frac{360\,N}{\frac{28\alpha}{25}}\right)^2 \sin 360x \text{ deg.}$$

$$= 5.009\left(\frac{360\,N}{\alpha}\right)^2 \sin 360x \text{ deg.} \qquad (17)$$

A comparison of the acceleration curves for the three cams analyzed, as shown in Fig. 4, indicates clearly that the combination cam has the decided advantage in the second half of the cycle.

Constant Speed Cam

The same method as used to determine the form of a cam having a certain definite acceleration curve can be applied to other problems. For example, let it be assumed that the requirement is to maintain a constant speed during a part of the cam movement, as in a swinging knife with extremely long cutting line. A cleanly cut edge with longer intervals between sharpening of the knife can be maintained if the speed is as high as possible during the actual cutting. In Fig. 5 is shown the velocity curve of such a cam. The first derivative of its equation gives the acceleration and the integral gives the form of the cam.

Eccentric Cam

If the radius of the crankpin is larger than the crank radius, the result will be an eccentric, that is, a circular disk whose center of rotation is at a distance from the center of the disk equal to the crank radius. The throw of such an eccentric is equal to twice the eccentricity f and the working period as indicated in Fig. 6, is always 180 deg.

For the sake of simplicity, the influence of the dimensions of the cam follower is not taken into consideration. Then, when the cam rotates from its zero position to a certain angle α as indicated in Fig. 6, the cam follower will travel toward the center of rotation O, a distance S.

$$S = (f + r) - R$$

As derived in textbooks, this equation takes the form:

$$S = \frac{F}{2}\left(1 + \frac{r}{f} - \cos\alpha - \sqrt{\cos^2\alpha + C}\right) \qquad (18)$$

In this equation, $$C = \left(\frac{r}{f}\right)^2 - 1$$

If the speed of rotation of cam is N revolutions per second or $2\pi N$ radians per second; the time necessary for the rotation of the cam through an angle α will be:

$$t = \frac{\alpha}{2\pi N} \quad \text{or,} \quad \alpha = 2\pi N t \qquad (19)$$

Substituting this value of α in the above equation for S, and differentiating with respect to t, we obtain the equation for the speed of the cam follower, which will be:

$$v = 2\pi N\frac{F}{2}\left(\sin\alpha + \frac{\sin 2\alpha}{2\sqrt{\cos^2\alpha + C}}\right) \qquad (20)$$

The differentiation of this question with respect to t will give the equation for the acceleration, which will be:

$$a = (2\pi N)^2\frac{F}{2}\left[\cos\alpha + \frac{\cos 2\alpha}{2\sqrt{\cos^2\alpha + C}} - \frac{\sin^2 2\alpha}{4\sqrt{(\cos^2\alpha + C)^3}}\right] \qquad (21)$$

A theoretical limit for the eccentric cam is when the radius of the cam disk is infinitely large in comparison with the eccentricity of the cam. Under such conditions we have:

$$C = \left(\frac{r}{f}\right)^2 - 1 = \text{infinity}$$

Therefore, for these conditions, in the equations for speed and acceleration all members having C in the denominator can be considered as being equal to zero. This will reduce the above equations to:

$$v = 2\pi N\frac{F}{2}\sin\alpha \qquad (22)$$

$$a = (2\pi N)^2\frac{F}{2}\cos\alpha \qquad (23)$$

It will be noticed that if α be replaced by 180 deg. in Equations (9) and (10) as given for the Scotch Yoke mechanism, they will have the same form as the equations for the eccentric disk when r/f is equal to infinity.

Clearance Hole Tolerances for Interchangeability

A simple method for determining minimum hole clearances required for bolts and studs to assure interchangeable assembly without reaming

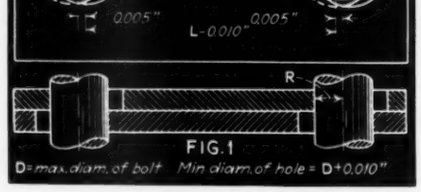
ROBERT PASH
Boesch Manufacturing Company, Inc.

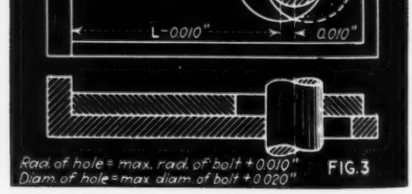
IN DECIDING upon the clearance to be used for a bolt hole, the general practice is to proportion it to the screw or bolt diameter; the larger the screw the greater the clearance. There is no uniformity in the amount of clearance assigned to each screw size. For a $\frac{1}{2}$-in. screw, for instance, different engineers will use from $\frac{1}{64}$ in. to $\frac{5}{64}$ in. clearance. The purpose of this article is to establish a logical basis for the correct selection of clearance hole sizes.

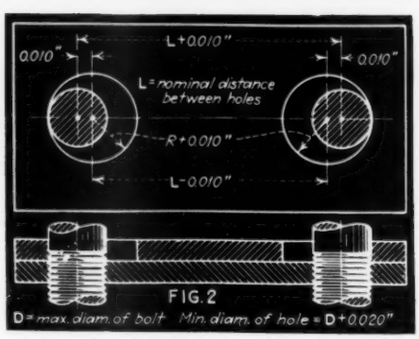
Clearance is defined as the difference in diameters between the body of a bolt or capscrew and the hole through which it is inserted.

If two plates are clamped together and drilled, a $\frac{1}{2}$-in. drill hole will allow the insertion of a $\frac{1}{2}$-in. bolt. Why then the clearance? The answer, of course, is obvious. Since it is impossible to have two sets of independently located holes register exactly, the holes must be made larger to allow the bolts to pass through. From this it follows that the required clearance is not related to the bolt diameter, but is dependent upon the accuracy in locating the holes.

The simplest case is that of two pairs of holes with through bolts. To illustrate, the distance between the hole centers in each part is assumed to be $L + 0.010$ in. The extreme case is shown in Fig. 1, from which it can be seen that the required clearance is 0.010 in. Ex-







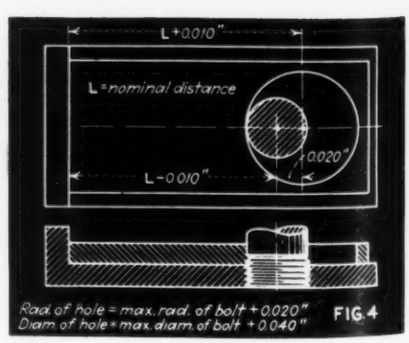


Fig. 1—For two pairs of holes for through bolts, minimum hole clearance must equal maximum tolerance for the center distance between holes. Fig. 2—If the two holes of one pair are tapped, clearance for the untapped holes must be twice the maximum tolerance for the center between holes. Fig. 3—Method of deriving required clearance for holes for through bolts when there is a shoulder fit between connected parts. Fig. 4—Method of deriving required clearance when a stud is used for holding connected parts

pressed in general terms, where two pairs of holes for through bolts are involved, the minimum clearance required in each hole must be equal to the tolerance on the distance between the hole centers.

Considering two pairs of holes, one set being tapped, the play of the screw in the tapped hole can be disregarded. The extreme condition is shown in Fig. 2. The clearance required in the untapped holes is 0.020 in., or twice that when none of the holes are tapped.

Another interesting example is when a shoulder is used to locate a part. The required clearances are derived as indicated in Figs. 3 and 4.

To establish the limiting conditions for the passage of a bolt through two holes eccentrically situated, from Fig. 5 it is seen that:

$$\frac{H_1}{2} = E_1 + \frac{D}{2} \qquad H_1 - D = 2E_1 \qquad (1)$$

Likewise, $$H_2 - D = 2E_2 \qquad (2)$$

Adding Equations (1) and (2)

$$(H_1 - D) + (H_2 - D) = 2E_1 + 2E_2 = 2(E_1 + E_2)$$

But, $$E_1 + E_2 = S$$

Hence, $$(H_1 - D) + (H_2 - D) = 2S \qquad (3)$$

This equation can be expressed as follows:

To be able to insert a bolt through two holes the sum of the clearances of the two holes must be more than twice the distance between the hole centers.

To establish clearances that will take care of all conditions, we must consider the maximum possible distance between any two mating holes. There is no need of having the two mating holes of the same diameter. One hole might be a close fit and all the required clearance furnished by its mate. That is exactly the condition where a tapped hole is used. To be consistent, the untapped hole used with a capscrew should have twice the clearance of the holes used with a through bolt. This can also be seen in the examples shown in Figs. 2 and 4 and the relationships noted thereon.

Consider the problem in Fig. 6. The extreme possible distance between any two pairs of mating holes must be found. Dimensioning on rectangular coordinates has been used purposely since it is the only way

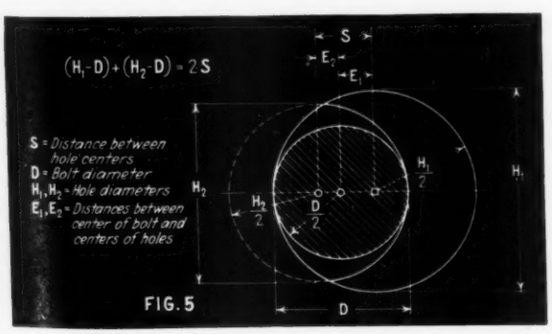

(H₁-D)+(H₂-D)=2S
S = Distance between hole centers
D = Bolt diameter
H₁,H₂ = Hole diameters
E₁,E₂ = Distances between center of bolt and centers of holes
FIG.5

Fig. 5—Derivation of the relationship required in order that a bolt can be inserted through two contiguous holes

to indicate dimensions that will definitely limit the possible distance between any two holes.

In Fig. 7 the tolerance on the location of one hole is T_1 and that of its mate is T_2. The center of the first hole might be anywhere in the larger square, its mate anywhere in the smaller square. Obviously, the greatest possible distance between any two points respectively in each square is from the corner of one square to the diagonally opposite corner of the other square. That distance is $(T_1 + T_2)\sqrt{2}$. From the statement of Equation (3) we derive:

$$\text{Total clearance} = 2(T_1 + T_2)\sqrt{2}$$

Referring to Fig. 6:

$$T_1 = T_2 = 0.010 \text{ in.}$$

$$\text{Total clearance} = 2(0.02)\sqrt{2} = 0.056 \text{ in.}$$

Dividing this equally between each hole of a pair, for bolts of 0.750 in. maximum diameter,

$$\text{Hole Diameter} = 0.750 + 0.028 = 0.778 \text{ in. min.}$$

If the holes in one plate were tapped, the diameter of the holes in the other plate would be:

$$\text{Hole Diameter} = 0.750 + 0.056 = 0.806 \text{ in. min.}$$

This equation assumes that tapped holes would be located as accurately as plain ones. To allow for eccentricity in tapping, still larger celarances will be required. Table I has been computed for the clearance to be given to each hole when located as in Fig. 6. The tolerances apply to base line dimensioning or rectangular coordinates.

Where each of two parts have two lines at right

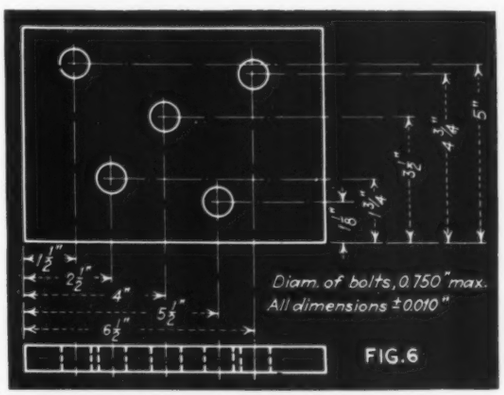


Fig. 6—Illustrating the common problem wherein two parts are to be machined independently and it is required to specify the minimum diameter of the bolt holes to allow assembly without reaming

angles that must coincide at assembly, it is necessary that those lines be used as reference axes for the location of holes. Otherwise exact alignment might be impossible, even though all the bolts or screws could be inserted.

In this discussion, reference has always been made to two parts. The reasoning, however, applies equally

well to any number of superimposed parts provided the center lines of the holes are parallel.

Application

The relation derived here can be applied on large and small production. Whenever true interchangeability is to be secured—that is, interchangeability independent of any set of jigs, accuracy of location must bear the correct relation to screw or bolt clearance. Even on special machines "spotting through" is no longer considered good practice. If small clearances are used the chances are that, in the replacement of the part, the holes will have to be "spotted back" by using hardened pointed screws and pounding on the part to be drilled. Sometimes even that is not possible. The proper way is to lay out and drill each part entirely independently. The tolerance on the location of the holes should be such as can be readily met by the means (hand layout, milling machine, jig border) to be used in the shop. The hole clearances should then be specified large enough to allow assembly under all conditions.

When the most accurate location is desired by the screws or bolts alone, the logical way is to have two hole-centers of each part determine one of the reference lines (one hole-center being the origin). The screw at the origin can then be a close fit, while all the others will need clearances in keeping with their accuracy of location from the reference lines.

Except as just mentioned or where unusual requirements exist, such as to avoid weakening the part, the writer can see no reason why the limit on clearance holes should not be so large that the hole would show on the side of the screw head when it is off the hole center as far as it will go.

Interchangeable Manufacture

Consider quantity production where drill jigs are used. If the same jig is used for mating parts, as is often done, those holes will match regardless of whether or not the jig is accurate. That, however, is not true interchangeability, for should a new jig be built from the drawings the chances would be that parts made from the second jig would not assemble with those made from the first. True interchangeability must depend on records alone. To secure it the relation derived herein must exist.

The relation derived is also of value in the use of parts or devices bought from other concerns. If the supplier will furnish the size of clearance holes used and the accuracy of their location, it will allow the machine builder to provide the mating holes in advance at minimum cost, allowing him to use the largest permissible location tolerances.

There is often no good reason for using the same accuracy of hole location on both mating parts. High accuracy in a part made in quantity allows less accuracy to be used on the mating part, the holes of which might have to be laid out by hand. The clear understanding of the relation between clearance and accuracy of location of holes can, no doubt, bring about many advantages and economies in manufacturing.

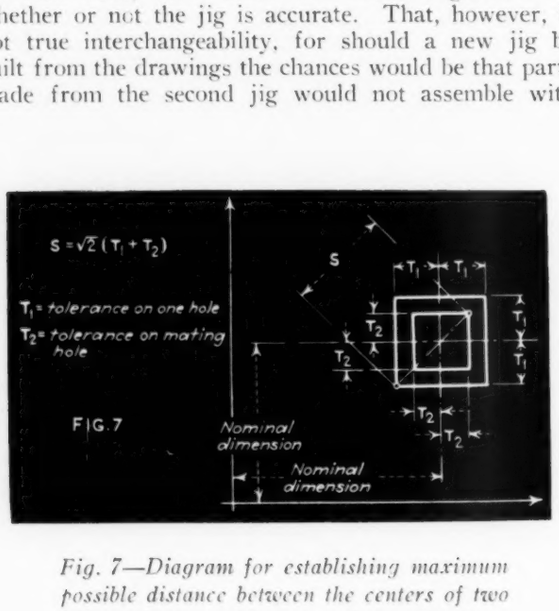


FIG. 7

Fig. 7—Diagram for establishing maximum possible distance between the centers of two holes whose positions are dimensioned on rectangular coordinates

Summary

For the same accuracy of hole location, capscrews and studs will require holes with twice as much clearance as required for through bolts.

Two pairs of holes for through bolts require clearance equal to the tolerance each way on the distance between the hole centers.

Total clearance for any pair of matching holes dimensioned on rectangular coordinates is determined by the formula:

$$\text{Total clearance} = 2\,(T_1 - T_2)\,\sqrt{2}$$

Where two lines on the mating parts must be in a definite relationship, the holes must be dimensioned from those lines.

Standardizing accuracy of hole location permits standardization of corresponding hole clearances by the above formula. This insures assembly of parts under all conditions.

Table I—Clearances to be Allowed for Holes Located by Rectangular Coordinates

Clearance added to maximum bolt diameter will give minimum hole diameter.
Tolerances are plus and minus.
The tolerances on the location of tapped holes have been selected larger because of the possible additional error in tapping the hole.

Clearance for Through Bolts		Clearance for Bolts in Tapped Holes		
		Tolerance on Location		
Tolerance on Location	Clearance for Each Hole	Untapped Hole	Tapped Hole	Clearance for Untapped Hole
0.001	0.003	0.001	0.002	0.009
0.002	0.006	0.002	0.003	0.014
0.003	0.009	0.003	0.005	0.023
0.004	0.011	0.004	0.006	0.028
0.005	0.014	0.005	0.007	0.034
0.006	0.017	0.006	0.009	0.043
0.007	0.020	0.007	0.010	0.048
0.008	0.023	0.008	0.012	0.057
0.009	0.026	0.009	0.014	0.065
0.010	0.028	0.010	0.015	0.071
0.011	0.031	0.011	0.017	0.079
0.012	0.034	0.012	0.018	0.085
0.013	0.037	0.013	0.020	0.094
0.014	0.040	0.014	0.021	0.099
0.015	0.043	0.015	0.023	0.108

Cold-Drawn Steel for Special Sections

When parts cut from special steel sections satisfy the design requirements, properly designed cold-drawn sections offer advantages in simplified design, longer wear, closer tolerances, and improved appearance as well as reduced machining costs

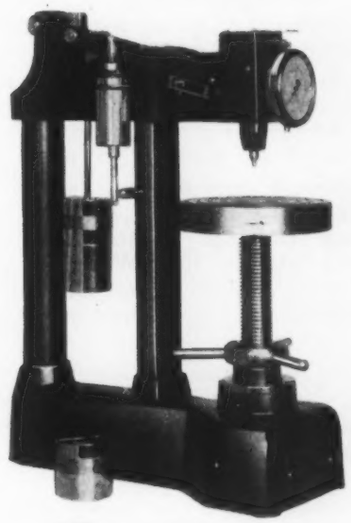


Fig. 1—In the hardness testing machine made by Wilson Mechanical Instrument Co. drawn steel pillars support the head

WHEN a given cross section is desired along all or a major portion of one axis, steel parts are usually made from bar stock, a casting or forging, or a cold-drawn section. A two-fold design problem is involved in selecting a special cold-drawn section embodying this prevalent cross section. The primary advantage in the use of special drawn sections is reduced machining cost; but first, the requirements of the design must be satisfied by the cold-drawn materials available and their physical properties, considering the limitations of the process and the quantity needed. In adapting the shape of the part to the cold-drawing process—the second phase of the design problem—many other advantages may be obtained such as improved appearance, simplified design, increased strength and longer wear, in addition to the reduced cost of machining.

All grades of carbon, alloy or stainless steel that can be hot-rolled are readily obtainable in cold-drawn form, although the softer steels, those low in carbon and hardening alloys, yield most readily to the cold-drawing process. While a soft steel may be drawn to a comparatively simple cross section in two or three passes through progressive dies, a harder steel may require additional passes and possibly preliminary and intermediate annealing treatment to produce the same section.

Since the drawing operation adds greatly to the strength of the steel, these softer cold-drawn steels provide the strength obtained in the harder grades of cast, forged or hot-rolled steels. For this reason, low-carbon and low-alloy steels listed in the table may be safely specified in the cold-drawn condition where hot-rolled steel of higher carbon or alloy content might be necessary. Cold-drawing often can thus be employed as a substitute for heat-treatment.

Tensile strength, yield point, and hardness in any of these steels are increased by cold-drawing, while the ductility is decreased. In producing cold-drawn sections, the soft hot-rolled bars are pulled through dies which reduce the area of the cross section from 10 to 15 per cent, depending on the size of the section and grade of steel.

Although machinability of most carbon and alloy steels is improved by cold-drawing, some steels are inherently more easily machined than others, and this property has been highly developed in certain grades of "free-machining" steels. Where operations such as cutting off, drilling, tapping or finish milling are involved, ease of machining is a factor of importance in selecting the material and the fabricating process. Conservative values of machinability are also included in the table for this list of cold-drawn steels.

In addition to increasing the machinability and strength, the cold-drawing operation produces a smooth, bright finish—an excellent surface for appearance. Lacquer, paint or enamel can be applied directly to the cold-drawn section with no surface preparation. This

is one of the factors in the selection of cold-drawn stock for the pillars of the Rockwell hardness tester shown in Fig. 1. But the desirable surface finish is incidental to other improvements made possible in this application by the special section, such as increased throat depth without a corresponding increase in the overhang of the head.

Cold-Worked Surface Resists Wear

Elimination of expensive form-milling operations is only one advantage in using a cold-drawn section shown in Fig. 2, for Otis elevator door rails. Another factor in this application is that tolerances on cold-drawn sections are much smaller than the dimensional variations in castings, forging or hot-rolled sections. As compared with $\pm \frac{1}{16}$ in. for hot-rolled forms, commercial tolerances for cold-drawn sections are only $\pm$ 0.003 in. In fact, the dimensional and angle tolerances readily obtainable in cold-drawn sections are representative of good machining practice. This factor makes it possible to take advantage of the hardness and wear-resistance introduced by the cold-drawing operation. Under the rolling action of the wheels supporting the elevator door, the cold-worked rail surface wears much longer than would a hot-rolled steel rail.

In these special sections metal is not removed in the cold-drawing operation but is redistributed without loss of material. Thus, metal cost as well as machining expense is reduced, while further cost advantages arise from the rapid rate at which the drawing operation is performed, production time in practically every instance being much less than for machining.

While it is preferable to begin the cold-drawing operation with a hot-rolled shape closely approximating the cross-section desired, the cold-drawing process has been developed to the point where intricate sections can be drawn from unrelated hot-rolled sections such as rounds, squares and flats. Some suppliers who produce large quantities of these special sections have small hot-rolling mills for "breaking down" standard bars to special shapes more suitable for cold drawing. With this hot-rolling equipment, sections with internal and external undercuts such as shown in Fig. 3 can be prepared from standard shapes. However, the closer a drawn section approximates available hot-rolled sections, the greater will be the opportunity for cost saving and prompt delivery.

But there are practical limitations in the shapes obtainable by the cold-drawing process. The metal in all parts of the section must have sufficient strength to pull itself through the die, thus limiting the possible amount of reduction in area and cold working. Although the metal can be made to flow into extended sections of the die, the amount of this redistribution is limited by the strength of the bar leaving the die.

Approximate Physical Properties of Cold-Drawn Steels

	Tensile Strength lb. per sq.in.	Yield Point lb. per sq.in.	Elongation per cent	Brinell Hardness	Machinability Surface Feet per min.
SAE-1010	60/70,000	55/65,000	20/30	137/149	105
SAE-1020	70/85,000	60/70,000	15/25	149/170	115
SAE-1030	75/95,000	65/75,000	15/25	170/187	100
SAE-1040	80/100,000	70/80,000	14/20	187/202	85
SAE-1050	85/110,000	75/85,000	14/20	202/228	80
SAE-1112	80/95,000	70/80,000	15/25	170/196	150
SAE-1120	70/85,000	60/75,000	15/25	137/166	135
High Sulphur Bessemer Screw Steel	80/95,000	70/80,000	15/25	170/196	200
High Manganese Open Hearth Steel	75/85,000	55/65,000	15/25	143/170	140
X-1315	75/90,000	55/70,000	15/25	143/179	140
1020-90	75/90,000	65/75,000	15/25	170/187	130
Special Carburizing Steel	75/90,000	65/75,000	15/25	170/187	130
.30/.40 Carbon High Manganese Open Hearth Steel	85/95,000	70/80,000	15/25	187/207	115

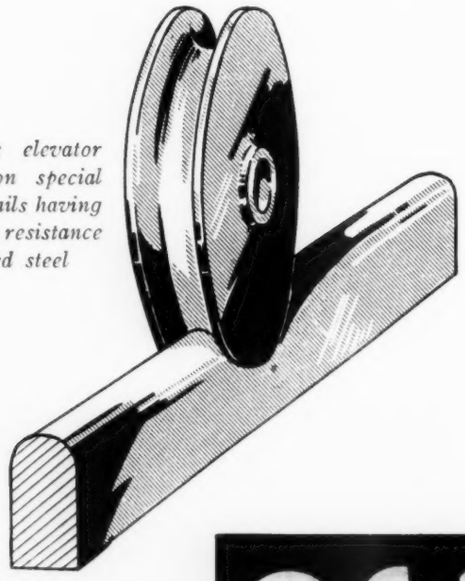


Fig. 2—Otis elevator doors roll on special drawn steel rails having greater wear resistance than hot-rolled steel

Courtesy Union Drawn Steel Company

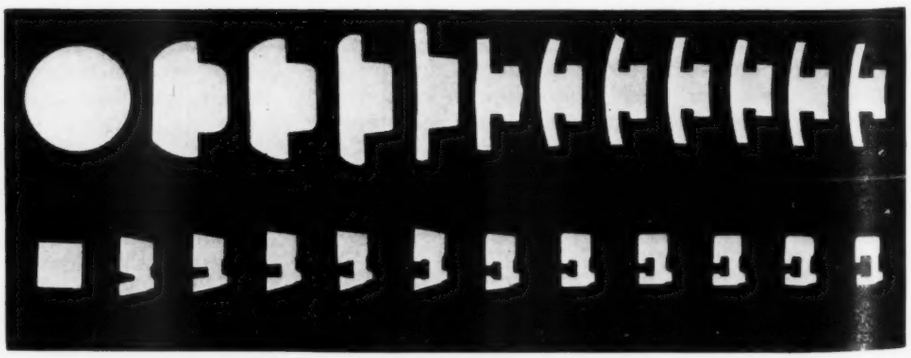


Fig. 3—(Above) Six hot-rolling passes and five cold-drawing passes and (below) one hot-rolling and ten cold-drawing passes are required to develop these special drawn sections

In the permanent magnet polepiece shown in Fig. 4 this is the limiting factor.

As shown at (*A*) in Fig. 4, the extensions are not strong enough to permit the metal to be pulled through the die. If the shape shown at (*B*) can be used, a cold-drawn section is possible providing a fillet radius is permissible at the pointed ends. If the design allows the use of a full radius, as shown at (*C*), the section will be more satisfactory to draw. However, the final decision in regard to the possibility of drawing such shapes rests with the steel supplier, as much depends on other factors such as equipment available and the experience of the manufacturer.

Sharp Corners in Drawn Sections

Reentrant or inside corners in some types of sections cannot be obtained satisfactorily with a radius less than $\frac{1}{64}$ in. because of excessive die wear. If a sharper corner is needed, an additional machining operation may be required to clean out the corners. And if an added machining operation is necessary, a generous radius should be used as the dies will then last longer. On the other hand, sharp external angles and corners are readily obtainable on cold-drawn sections, this being a feature in which the cold-drawing process excels hot-rolling, casting or forging.

If the requirements of the design are satisfied by materials available in the cold-drawn state, and if the shape desired can be produced by cold drawing, the remaining factor is the quantity needed.

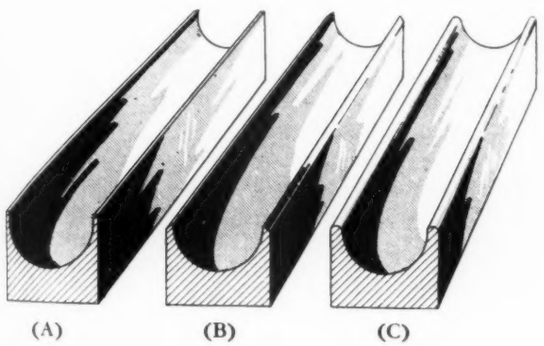


Fig. 4—Extensions of the section (A) *may not have sufficient strength for drawing. Section shown at* (B) *can be drawn, but* (C) *eliminates undesirable sharp corners and thin sections*

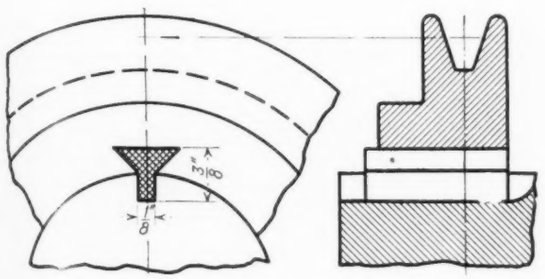


Fig. 5—Small added cost in this special key section is of little importance in comparison with the reduction in keyway pressure

Frequently in the use of cold-drawn steel sections other advantages will outweigh that of economy. A few of these, illustrated in this article, are:

1. A surface that needs no preparation for plated or painted finish.

2. A hard skin that is highly wear resistant.

3. Shapes obtainable in one piece where otherwise an assembly would be required.

4. Simplification of associated parts made possible by irregular shaped drawn sections.

5. Flexibility in designing because irregular sections involve no added cost.

Material cost in cold-drawn sections, as in either special hot-rolled or standard bar stock, varies with the quantity, and for this reason quantity is a critical factor in unit cost per finished piece. In finished parts from special drawn sections three elements enter into the cost of the piece: Cost of the hot-rolled section (ready for the drawing operation), die expense and drawing cost, and the amount of finish machining.

Factors Affecting Cost

In the cost of the hot-rolled bars there are also three factors: grade of steel, quantity and shape. If the special sections can be drawn from standard hot-rolled sections (squares, hexagons, rounds or flats) established prices will prevail for the hot-rolled bars.

For special hot-rolled bars, such as used in developing the sections shown in Fig. 3, the cost will be governed largely by the intricacy of the shape and the number of rolls and rolling passes required. Most suppliers have roll equipment available for hundreds of sections; but, if special rolls are required, their cost will be added to the cost of the finished section.

Cost figures for the cold drawing dies vary with the size and design as well as the grade of steel specified, and to some extent with the tolerances allowed. But no reliable cost data can be given here, as much depends on the experience and ingenuity of the supplier, and the equipment already available. For small quantities, the die cost will usually be the largest item in the total cost of the initial order.

For quantities of a ton or more for the initial order and similar quantities as frequently as once a year, die costs will be a comparatively small part of the cost per piece. With sufficient quantity—five or ten tons for example—special sections may cost less than even simple machining operations on a standard bar. A small machine-tool spindle consisting of a standard round with a small flat to prevent rotation of the spindle,

when made by cutting a special drawn section to length, not only costs less than the equivalent machined bar, but the close tolerances and the bright surface make all surface-finish machining unnecessary.

Where sufficient quantity is required, the greatest advantage in the use of special drawn sections is the possible flexibility in design. Sections such as those shown in Fig. 3, impossible to obtain by any other means, often result in unusual savings in cost and improved performance of the equipment. But complicated shape alone is not the only advantage. For

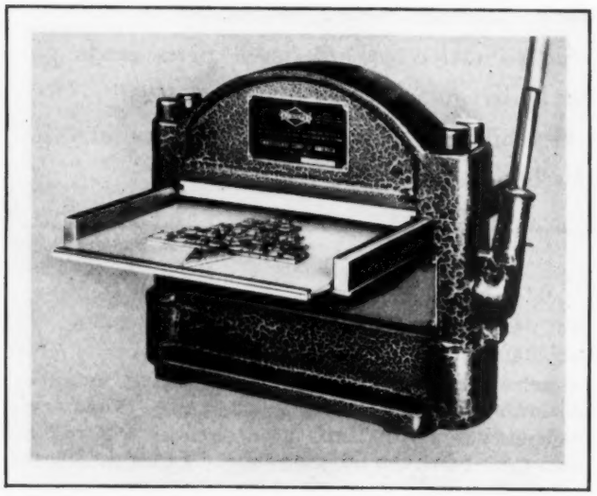
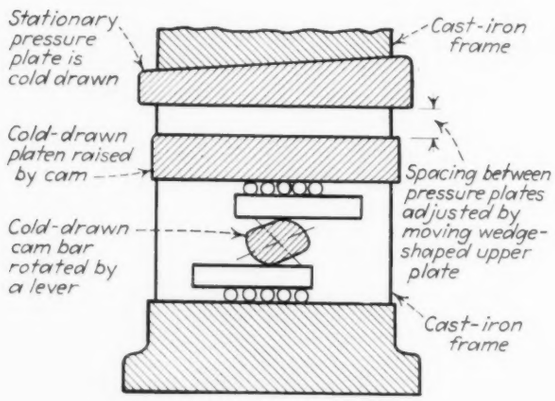


Fig. 6—Special drawn steel sections in the Embosograf reduces machining costs and also simplified the design. Fig. 7—Rotation of the cam lifts the lower plate against the adjustable upper plate, as shown in the section view below


Stationary pressure plate is cold drawn
Cast-iron frame
Cold-drawn platen raised by cam
Spacing between pressure plates adjusted by moving wedge-shaped upper plate
Cold-drawn cam bar rotated by a lever
Cast-iron frame

instance, a simple key section shown in Fig. 5, makes possible the use of die-cast pulleys in small machine tools. The flared head of this key section distributes the key pressure over an extended area, preventing enlargement of the keyway in the soft die-casting. In addition to distributing the driving force, this key places the metal in the hub between the key head and shaft in compression, tending to lock the pulley to the shaft. Both of these advantages are obtained in a high-strength, special drawn section at little extra cost.

Special sections simplify design and improve performance as well as reduce cost in the embossing machine illustrated in Fig. 6. A cam bar, cold-drawn to final shape, develops the necessary pressure when rolled between the bearing plates, as shown in Fig. 7. By any machining operation this cam contour is expensive; with cold-drawn stock any desirable contour is obtainable. At the same time, the greater surface hardness of the cold drawn material forms a more wear-resistant surface and the added strength of the cold-worked steel makes possible the use of a less expensive material.

Another cold-drawn section in this machine, the lower pressure plate, has large radii on the two upper corners, provided at small cost. In the upper pressure plate, also cold drawn, the angle between the two faces is obtained without the expense of removing any metal, and provides a simple means for adjusting the spacing between the two pressure plates. Again, large radii needed at the corners involve no additional cost, and of course, no finish machining is necessary on the drawn surface.

Simplification in the design of a permanent magnet (also illustrated on page 162 in May *Product Engineering*) is made possible by using a cold-drawn section for the polepiece, as shown in Fig. 8. To machine this part from bar stock involves prohibitive machining expense. In the cold-drawn section, even the projection required for resistance welding the polepiece to the permanent magnet is obtained at no appreciable added cost.

Minimum tolerances of ±0.003 in. in this special section are too large for the cylindrical surfaces forming the air gap in which the instrument rotates. However, a light grinding operation, removing less than 0.005 in. of metal, is the only additional operation required to assure a tolerance of ±0.001 in. on this surface.

Many other examples are available demonstrating successful applications of special drawn steel sections. But the success in each instance depends upon the care used in adapting the design to the cold drawing process after considering the limitations and advantages of the method.

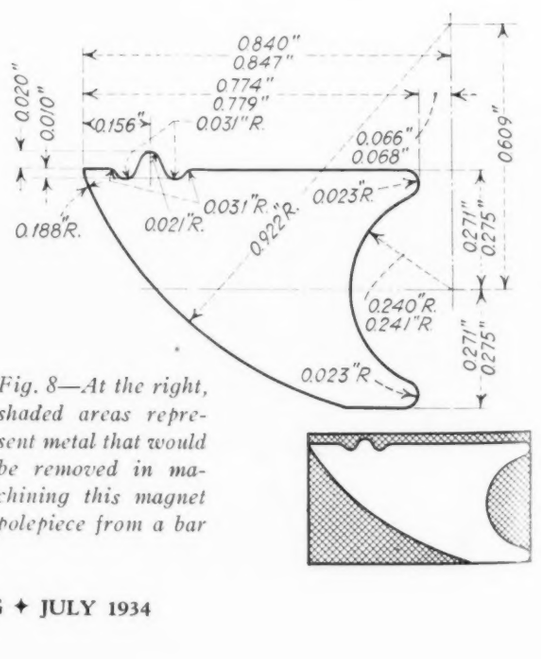


Fig. 8—At the right, shaded areas represent metal that would be removed in machining this magnet polepiece from a bar

Spheres for Joining Tubes

Hollow steel connecting spheres are used in a new type of arc-welded structure described in a paper submitted in the second Lincoln Arc Welding Prize Competition

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HAROLD KENNEY
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TUBES are unique among all sections in that for a given weight of material they provide not only the greatest section modulus but equal strength in all directions. They are thus the ideal sections for columns, and are also suitable for tension members because of their equal resistance to lateral movement in all directions.

But the chief difficulty in the use of tubular members is in attaching or joining the parts. In one method the tubes are slotted and welded to a connecting plate which passes through the axis of each tube, as illustrated in Fig. 2; but this method is expensive. The usual method in airplane construction, even more costly, is to weld the tubes directly to one another, having previously shaped them to fit.

In addition to being expensive such welds are difficult to inspect to determine if the welding has been performed satisfactorily, particularly where the tube is welded to a flat member at an angle with the tube axis, as shown in Fig. 3. Also when several tubes are assembled together into a single joint there is a further disadvantage in that the metal is subjected to excessive overheating in making the successive welds.

To overcome the difficulties in the use of tubes, the tubular members can be joined, as shown in Fig. 4, by welding them to a common spherical member. Each of the tubes is joined at right angles, the line of contact being a circle. Thus, it is possible to reduce the welding to a single type of the simplest kind. For lightness the sphere is usually hollow, and may be fabricated from scrap plate by welding. The most convenient procedure is to weld together two die-pressed hemispheres.

Spherical joints reduce the expense of prepar-

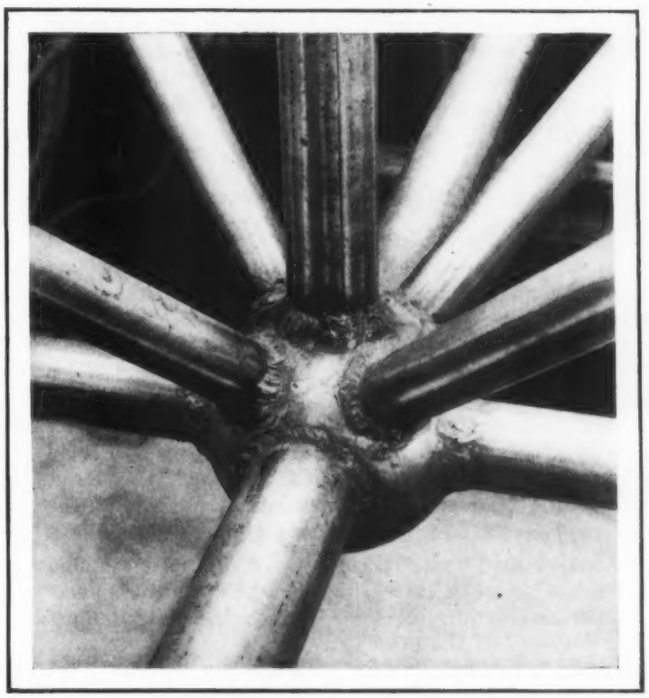
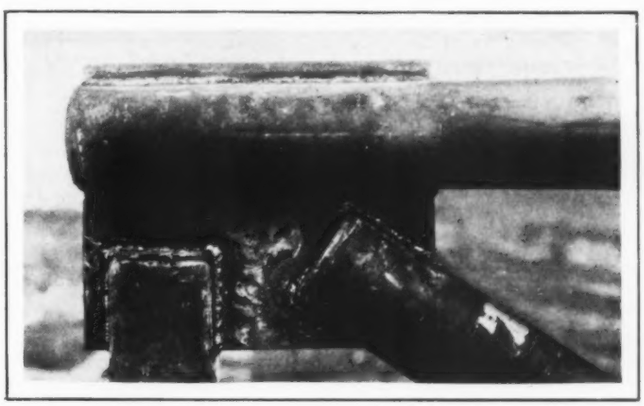


Fig. 1 (Above)—Tubular members welded into spheres at the joints provide an economical, light, safe and strong structure. Fig. 2—(Below)—Slotted tubes welded to a connecting plate form a satisfactory but expensive joint



ing the ends of the tube, requiring only that the tubes be cut off square. A beveling operation is desirable on tubes of more than $\frac{1}{8}$ in. in wall thickness, in order to obtain better penetration of the weld. This beveling, however, is a simple and inexpensive operation, and both the cutting and beveling can be done on a mass production scale. Since the section of the weld between the tube and sphere is always the same, the welding operation itself and the subsequent gaging of the weld are both simplified.

When spheres are used, eccentricity at the joint can be prevented readily. If the tube is touching the sphere at all points, then the center line of the tube must pass through the center of the sphere. When the diameter of the sphere is large enough to accommodate all of the tubes in a cluster joint, as shown in Fig. 1, the welds can be made on the surface of the sphere without excessive overheating.

A further advantage of the spherical joint is that there is no sudden change of direction of the lines of stress. There is no stress concentration at the joint, as is common with tubes that are welded directly to one another.

Design of the Spheres

The method of welding adopted for the center weld between the two halves of the hemisphere will depend upon the number required, but as large quantities will usually be needed for each individual size, resistence or flash welding is preferable. When resistance welded, the dimensions of each hemisphere prior to flash welding must provide extra metal for melting and upsetting in the flash-welding process. Locating holes are required in each hemisphere, so that they can be held correctly in the jaws of the welding machine. After the welding operation, it is necessary to machine off the outside bead or flash on the surface of the sphere.

Spheres used for structural work need not be solid to withstand the stresses imposed when used to connect comparatively thin tubes. On the other hand, there is a limited thickness below which the spheres would collapse. Spheres of 4 in. outside diameter with a thickness of $\frac{3}{16}$ in. are of adequate strength for the tubes with which they have so far been used. In general the spheres should be $1\frac{1}{2}$ times the diameter of the largest tube with which they are used, and the thickness of the sphere should be $1\frac{1}{2}$ times the wall thickness of any tube to which it is connected.

In designing tubular structures, the chief imperfections which occur are faults in manufacture, variations in modulus of elasticity, eccentric loading, indefinite type of end supports and initial stresses.

Higher Working Stress Allowable

Considering actual support at the ends, no joints, unless made with a perfectly machined pin, can be considered free, and no joint can be absolutely rigid. To provide an additional factor of safety, these tubular structures are designed as pin-jointed members, neglecting the degree of fixity of the end. To allow for initial stresses in the tube caused by cold straightening and other conditions arising from the manufacture of the tubes, the permissible working stresses are reduced by 15 per cent.

Equivalent eccentricity allowances take into account values determined by experience. By using the spherical joint, virtually perfect concentricity in loading is obtained and no corrections are necessary. Instead of the usual value of 18,000 lb. per sq.in., a stress of 31,400 lb. per sq.in. can be safely used with this type of construction, provided that all the loads are known and stresses are calculated carefully. No overloading is permissible, of course.

In comparing the total weight of identical designs assembled by angle welding, and by the tube and sphere design, the tubular structure will show a saving of nearly 40 per cent. At the same time, as shown above, this construction is stronger and can carry higher working stresses.

The greatest objection to the use of tubes as structural members is that the steel in the form of tubes costs more than in the form of angles, channels and plates. However, lighter sections can be used with tubes, and the absence of eccentric loading will reduce the sections required, so that the cost of steel in the tubular design is approximately only twice the cost of steel in an equivalent structure using rolled sections.

When fabricated from standard sections and plates, welding costs for identical designs will be approximately 70 per cent greater than for the tubular design—a saving of approximately 40 per cent in favor of the tubular design.

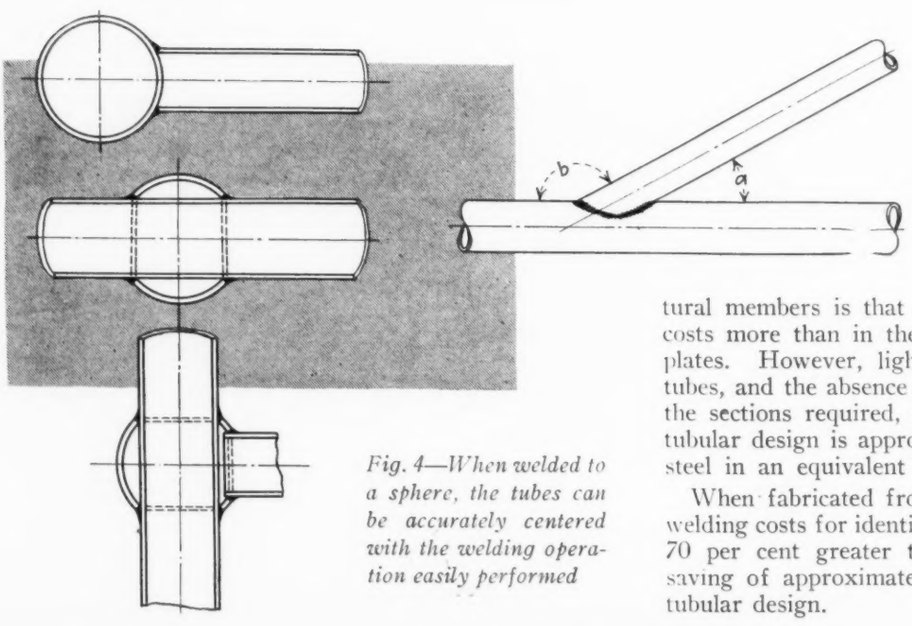


Fig. 3—Tubes joined to another tube or to a flat plate present a difficult welding problem when the angle is acute

Fig. 4—When welded to a sphere, the tubes can be accurately centered with the welding operation easily performed

Laminated • Sheet Bronze • Graphited
BEARINGS

Construction of and alloys used for the different types of bearings, the basis for the selection of the type of bearing to use, and examples of their typical applications

C. H. LEIS
Chief Engineer, Johnson Bronze Company

SHEET bronze bushings are manufactured from cold rolled bronze and their hardness is under complete control at the time of rolling. The cold rolling imparts a very dense structure, which gives to these bronzes certain bearing qualities that are not ordinarily obtained in cast bronzes. In large quantities, sheet bronze bushings cost considerably less than the cast bushings.

Certain types of machinery, because of low unit cost, compel the engineer to choose the most economical bearing material. There are numerous applications wherein the relative motion of the shaft is so slight that, even with continuous wear, the sheet bronze bushing will outlast the machine itself. This type of bushing is most extensively used in the automotive industry for this particular reason. Sheet bronze bushings are also ideal and economical for use where atmospheric or gaseous conditions might cause cast-iron bearings or steel bearings to corrode. This is especially true of machinery or motor parts that are subjected to various weather conditions, as in argicultural equipment. Similarly, factory skylight or ventilating windows must withstand rain, heat, and gases from within, all of which cause corrosion. In such places all movable parts should be equipped with sheet bronze bushings to insure positive and permanent operation.

The alloys used for this type of bushing are selected to serve specific conditions as set forth in the applications of the alloys described as follows:

Johnson Alloy No. 40. Chemical composition: copper 90.0, tin 0.5, zinc 9.5. Physical properties: Tensile strength 60,000 to 70,000 lb. per sq. in.; yield point 47,500 to 52,500 lb. per sq. in.; elongation per cent in 2 in. 5 to 15; hardness, per specifications.

Bushings made of this alloy function satisfactorily in many places as a result of the physical working of the alloy during its manufacture rather than by virtue of its chemical composition. It is used to replace cast bushings in many places where the impact load is the important factor and generally where the motion is oscillatory and not rotary.

In the automotive industry, bushings made of this alloy are used for rocker arms, steering knuckles, steering mechanisms, torque tubes, universal joints, spring eye bushings, starting motors, distributor shafts and places where rotary speed is not high. These bushings are also used in many places on the chassis frame and brake system. This alloy is extensively used for thrust washers. Bushings made of this alloy should be burnished in place as it has no machining qualities.

Johnson Alloy No. 44. Chemical composition; Copper 88.0, tin 4.0, zinc 4.0, lead 4.0. Physical properties: Tensile strength 70,000-80,000 lb.; yield point 50,000-60,000 lb. per sq.in.; elongation per cent in 2 in. 5 to 15; hardness, per specifications.

Bushings made of this alloy are recommended for heavy duty service. Alloy No. 44 withstands high impact loading as well as rotating motions. Graphited bushings made of this alloy perform well under comparatively high speeds. Bushings made of this alloy can be burnished, bored or reamed in place.

This alloy is recommended for piston pin bushings, differential gear bushings, transmission gear bushings, electric hoists, fractional horsepower motors, washing machines, household appliances and agricultural machinery.

Laminated Bearings

Laminated material is used to obtain a bearing that will resist high unit loads and have sufficient impact capacity. Such qualities can only be obtained by fusing

a highly efficient bearing material to the surface of a metal having physical properties in excess of the requirements.

Plastic elements are always of low compression strength. In order to obtain the desired qualifications such elements must be comparatively thin to prevent pounding out. Bearing elements of a plastic nature pound out easily. It is therefore important that the bearing material be thin. The best results are obtained if the thickness is held to a minimum, which is governed by the conditions under which the bearing must operate.

There are three distinct types of laminated bearings: (1) bronze-backed, babbitt-lined; (2) steel-backed, babbitt-lined; (3) steel-backed, bronze-lined.

Bronze-Backed, Babbitt-Lined Bearings

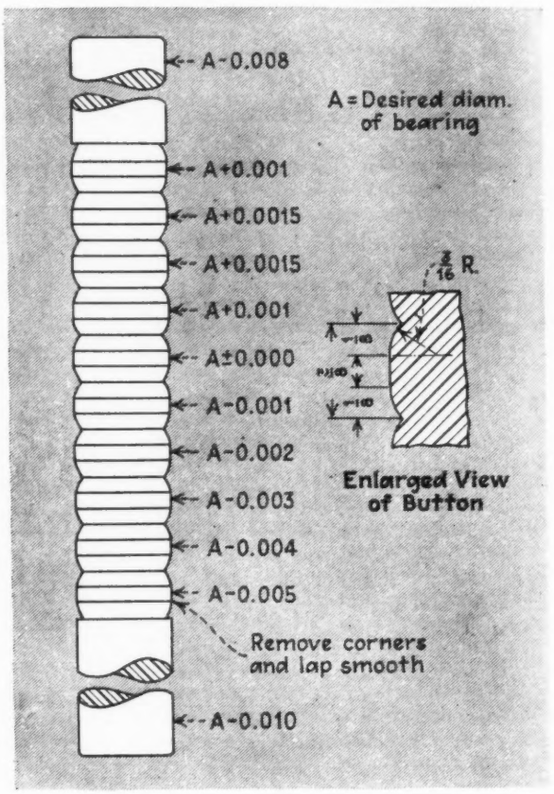
Until recently, bronze-backed, babbitt-lined bearings have been used in internal combustion engines as main bearings. The alloy used in the bronze backing was Johnson Bronze Alloy No. 66, (S. A. E. No. 66), and it was adopted chiefly for two reasons—good machining qualities and low cost. In the early stages of the automotive industry with comparatively light loads and slow speeds, the conditions under which this alloy functioned were such that it answered the purpose. But speeds and compressions were increased, placing extraordinary demands on the backing material, and because it lacked certain physical properties, many bearing failures occurred.

It was not generally recognized that such failures of main bearings were directly caused by the low pounding resistance of Alloy S.A.E. No. 66. In many instances a substitution of Alloy S.A.E. No. 63 resulted in the elimination of the failures as this alloy was capable of withstanding greater pounding at elevated temperatures. The failure of Alloy S.A.E. No. 66 naturally was followed by new developments which evolved the steel-backed and babbitt-lined bearing which had previously been used in the aircraft industry in Liberty motors.

The bronze-backed, babbitt-lined bearing has many advantages over the steel-backed bearings. They are as follows: (1) Bronze is more comformable in the housing than is steel; (2) Conductivity is higher; (3) Bronze does not convey bearing noises as readily as steel, therefore the motor is quieter; (4) When failure of a bearing does occur, the shaft will not score with bronze-backed bearings as it does with steel-backed bearings; (5) In small quantities, bronze-backed bearings are more economical; (6) Flanged bronze-backed bearings permit thrust loads being carried on their faces.

In early applications, babbitt linings were selected chiefly according to their hardness. It was generally agreed that the harder babbitts gave the best service. This theory has been exploded with the present methods for testing alloys, at not only room temperatures but also at the elevated temperatures to which the bearings are subjected in service. The babbitt alloy that was generally used at the inception of internal combustion engines was S.A.E. No. 24, which contained 84.0 tin, 7.0 copper and 9.0 antimony. As the speeds and loads increased, S.A.E. No. 11 containing 87.0 tin, 6.0 copper and 7.0 antimony replaced the former alloy. It was considerably softer, but acted much better under the higher temperatures encountered in internal combustion engines.

Because its physical properties were not maintained at the higher temperatures, S.A.E. No. 11 was replaced by S.A.E. No. 10 in a great many instances. At present, Johnson Bronze Alloy No. 9 answers the specifications for babbitt to be used for high speeds and loads where higher temperatures are encountered in modern design. The content of this alloy is 89.0 tin, 3.5 copper and 7.5 antimony. It has been definitely proved by tests that an 89 Sn-3.5 Cu-7.5 Sb babbitt is capable of withstanding higher impact loads and greater flexing than either S.A.E. No. 10 or S.A.E. No. 11 Babbitts.



Burnishing bar for sheet bronze bushings. It is made of tool steel, hardened, ground and polished

Steel-Backed, Babbitt-Lined Bearings

The present era created a call for the economical manufacture of babbitt-lined bearings, and methods for laminating a thin layer of babbitt to a strip of steel were developed. With exact temperature control and precautions to prevent oxidation, the babbitt is fused onto the steel strip in a continuous process creating a perfect bond. The material is manufactured in flat strips, which permit the stamping of oil grooves and oil holes of any design. The standard thickness of steel is 0.050 in. while the standard thickness of babbitt

is 0.020 in. Either of these dimensions may be varied slightly. The total thickness of the lamination will not vary more than 0.001 in. Main, camshaft, and connecting rod bearings are produced extensively from this material. Bearings of this type can be made either to be line reamed or made to precision standards. Bushings requiring babbitt linings are manufactured in the same manner. The advantage gained by using this laminated material is purely an economical one.

Steel-Backed, Bronze-Lined Bearings

The low physical properties of babbitt at elevated temperatures, made necessary further research for a more suitable material to accommodate increased speeds and higher temperatures. It was necessary to keep in mind the highly desirable qualities of plasticity and low coefficient of friction that babbitt offers. These requisites were retained in an alloy containing copper and lead. The latter is held in suspension by catalytic and other agents. This alloy is slightly harder than the hard babbitts and is more able to resist pounding. Being fused to strip steel, the bearing is rigid. The lead content in this alloy can be varried from 30 per cent to 40 per cent. The lower the lead, the harder is the bronze, the harder bronzes requiring increased hardness of the shafts. Being a mechanical mixture, the lead is diffused through the copper in a dendritic structure. More research remains to be done on the application of this type of bearing.

Graphited Bearings

Bronze bearings with a graphite compound inserted in the inner bearing surface are recommended for many applications. They are especially useful in installations where it is desired to eliminate the necessity of oiling, as when access to the bearings is difficult or oiling is likely to be neglected.

The graphited bearings described here are designed to provide a rubbing surface of alternate sections of bronze and a graphite lubricating material. From 40 to 45 per cent of the entire bearing surface provides graphite contact with the shaft. This is accomplished by cutting minute, dovetailed grooves, approximately $\frac{1}{32}$ in. in width into the bearing surface. The grooves cut in the metal are filled with a graphite composition which is compacted under high pressure by a patented process. The serrations or grooves are filled flush with the bearing surface, producing a uniform and smooth lubricant of constant density throughout.

In any good bearing material the bearing structure is composed of hard bearing crystals surrounded by a more plastic metal. Minutely spaced alternating metallic and graphited surfaces produce a structure similar to that found in preferred bearing alloys. Here the graphite structure surrounds the bearing metal in a similar manner. The minute serrations do not affect the load carrying capacity of the bearing. Definite advantages are gained by confining the lubricant in the minute serrations, because the graphite compound is not dislodged through deformation during assembly into the housing. The depth of the groove is such that it will permit a long bearing life.

The graphite lubricant used in this type of bearing is the result of much experimenting and a long series of tests to determine the following necessary characteristics: (1) low coefficient of friction; (2) necessary hardness; (3) tenacity; (4) insolubility in water, alcohol, or oil and its products; (5) absorption of lubricants.

Two distinct types of graphited bearings to meet varying service are available: (1) Graphited cast bronze; (2) graphited sheet bronze.

Graphited Cast Bronze

Graphited bushings of the cast alloy type have the dovetailed serrations cut at an angle of 7 deg. to the axis, in order that the shaft will be in contact with bronze and graphite surfaces at all times. This facilitates the reaming operation after the bushings are in place. The standard alloys given in the article, "Bronze Bearing Alloys," appearing in the June number, can be used for this type of bushing, depending upon the operating conditons. Under average service Alloy No. 27 will function efficiently.

Cast graphited bushings are recommended for constant running shafts having a limit of 7,500 in. lb. per min. In other words, if the product of the peripheral speed of the shaft per minute times the load per square inch of projected bearing area does not exceed 7,500, this type of bearing can safely be used without further lubrication.

Applications of these bearings include:

Washing machines	Concrete mixers
Office equipment machinery	Dairy machinery
Cigarette manufacturing equipment	Sump pumps
Textile machinery	Tractors
Food products machinery	Universal joints
Automobile parts	Trolley wheels
Canning machinery	Traveling ovens
Household electrical equipment	Textile machinery

Sheet Bronze Graphited Bushings

Coiled strips of sheet bronze have serrations cut longitudinally, and into them the graphite lubricant is forced under high pressure. Cutting the grooves rather than pressing or indenting insures an unfractured structure and maintains uniform hardness and strength. Indenting by pressure displaces the metal and tends toward rupture. The graphited strips are then baked under controlled temperatures. In forming the flat sheet into a cylindrical shape the grooves automatically develop into dovetails which hold the graphite permanently in place.

Where slow, intermittent and oscillatory rotation of the shaft occurs, sheet bronze graphited bushings can be applied efficiently. They are most economical in large quantity production for applications such as rocker arms, brake levers, guide rollers, casters, sheaves, shock-absorbers, and railroad car doors.

Graphited sheet bronze bushings cannot be reamed. They should be burnished in place, using a burnishing bar such as shown in the accompanying drawing.

Editor's Note—The next in this series of articles on bearing metals and the design of sleeve-type bearings will be on the subject of their lubrication and will appear in the August number. This will be followed by an article on the practical design of bearings and methods of making calculations.

QUESTION AND COMMENT

Wobble Motion for Vibrator

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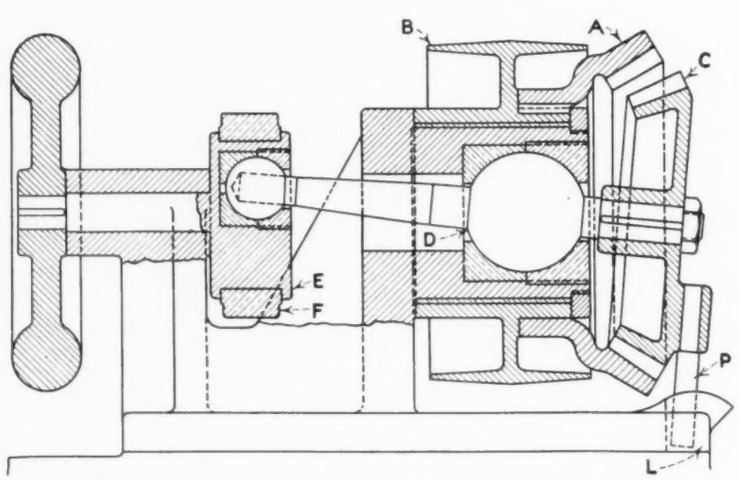
● To obtain a high-speed reciprocating motion from a slowly rotating power supply, the mechanism shown in the accompanying diagram operates on the reverse of the internal-gear overdrive principle, using relatively few parts.

Internal gear *A* is rotated by the drive pulley *B* and meshes with gear *C* which is held from rotating by pin *P* restrained in the slot *L*. However, the gear *C* can move in a circular path around the ball *D* as a center, rotating the eccentric *E* and imparting oscillation to *F*, an eccentric strap.

The velocity ratio between the gear *A* and the eccentric is readily obtained by considering power application in the reverse direction. With 42 and 40 teeth on gears *A* and *C* respectively, the gear *A* advances 2 teeth for each rotation of the eccentric. Therefore, the eccentric rotates 21 times for each revolution of the gear *A*. Or, with power supplied to the pulley *B*, the eccentric rotates 20 times for each revolution of the drive gear.

Gear teeth in this design are subjected to sliding action but the efficiency of the unit compares favorably with other types of gearing providing the same motion. The teeth are cut slightly deeper to compensate for the angular position of the inside gear as well as for the difference in the number of teeth in the gears.

When used to drive a vibrating table, inertia forces impose smaller loads and consequently less wear on the gear teeth when a flywheel is used on the eccentric shaft. The cast handwheel at the left in this design thus steadies the power required.



Using a non-rotating internal gear, a high-speed drive for a vibrating table requires but few parts

Design of Connecting-Rod End

ROLAND V. HUTCHINSON
Lansing, Mich.

● Detail design of the section surrounding the bushing in the connecting-rod end shown in the diagram depends upon fabrication methods as well as the applied loads. The part may be forged or welded, the fabricating method depending upon the shop facilities and upon overall dimensions of the complete part. Forging is assumed in the following analysis.

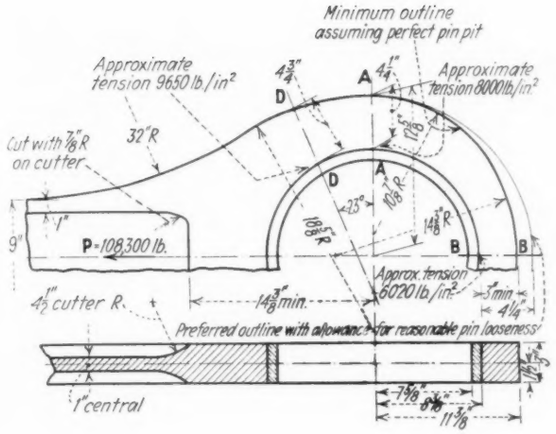
The rigid pin will limit deflection on the round side of the eye, forcing the eye to stay circular as external load is applied. Proportions of the section *B-B*, assuming the stresses to be circumferentially guided around the eye, will be affected by press fit conditions as well. For the dimensions *A* and the shank the proportions of the section *A-A* are governing factors. Under the external load, the pin does not control the extent of local deflection in this region; the eye is a bent beam of varying section eccentrically loaded.

Should more abrupt filleting than the approximate 32 in. radius be desired, the sections at both *A-A* and *B-B* will have to be deepened radially, for the change in fillet will increase the stress concentration around the hole.

In the sketch, it is assumed that the eye between *D* and the shank is rigid enough to limit deflections to the region between *A* and *D*. The outline is sketched in by the eye to reasonably meet this assumption, giving a section depth at *A* and *D* of $4\frac{1}{4}$ and $4\frac{3}{4}$ in., respectively. Bending moments and tension loads are taken as the product of half the shank load, the eccentricity, and the secant of 23 deg., the approximate angle between *A* and *D* scaled from the sketch. Eccentricity, bending moment and radical dimensions at *A* and *D* are inter-dependent, a fact that must be kept in mind in trial and error computation.

The circular bronze bushing will not remain circular when installed, the out-of-round condition reflecting the variation of radial pressure existing between the surfaces in contact. In the proportions given, the average radial pressure will not exceed 1,500 lb. per sq.in. and the average press-fit stress in the steel, at the inner wall of the eye bore, will be less than 4,000 lb. per sq.in. The bushing merely preloads the eye, and the effect can be neglected in practice as the external forces load it beyond the assembly stress range.

However, if a replacement bushing of original dimensions is installed on repair, and the pin is only rounded up a little (as frequently happens in rush repairs of heavy machinery) load distribution varies with the clearance. Loading is no longer distributed but is concentrated close to the section *B-B*. When the bearing surfaces do not conform, the eye can bend, requiring different proportions to meet the new conditions. Even initial clearance has this effect to some extent.


Minimum outline assuming perfect pin pit
Approximate tension 9650 lb/in²
Approximate tension 8000 lb/in²
Cut with 7/8" R on cutter
32" R
P=108,300 lb.
Approx. tension 6020 lb/in²
Preferred outline with allowance for reasonable pin looseness
1" central

Dimensions of the section of the bushed connecting-rod end depend on the bearing fit, the fillet radius and the assembly method.

For these reasons, it is desirable to increase the depth of section *B-B*, and for simplicity in manufacture, this section is made concentric with the pin center. But even with this design, if the bushing should disintegrate and the load come on the steel itself, the section at *B* will be greatly overstressed.

This example may well be the subject of photoelastic tests on bakelite models. Model tests could also be made using sheet rubber, ruled off in ¼ in. squares. From the rubber models good "horse-sense" pictures of the stress distribution may be obtained if only moderate loads are used in making the tests on the model.

Jewel Bearings in Electrical Instruments

To the Editor:

● With reference to the article "Jewel Bearings for Minimum Friction," appearing on page 178 of the May number, the usual electrical indicating instrument has a pivot of about 0.004 cm. radius, and carries a load of 2 to 5 gm. This pivot radius is about ⅓ of the radius recommended in the curve, Fig. 5 in the article.

In watt-hour meters, the pivot or ball radius is from 0.04 to 0.08 cm., and the bearing load is from 15 to 50 gm. These values of pivot radius are much larger than recommended in the curve.

Service experience shows that the pivot radii used in each of these two types of instruments are near the minimum. But the jewel bearings in indicating instruments are subject to intermittent service while in the watt-hour meter, the bearing is in continuous service. Therefore, the pivot radius recommended in the article should be modified according to the severity of the service.

The relation between torque and weight given in equation (6) in the article, works out well in practice. With the torque T in millimetergrams and the weight W in grams the ratio $T/W^{1.5}$ should not exceed 0.4 for electrical indicating instruments. This relation was checked against performance several years ago by plotting the ratio of torque to weight for a large number of instruments. The boundary line between satisfactory instruments and those with appreciable friction, gave the relation,

$$T/W^{1.33} = 0.375$$

Since this relation represents boundary line designs, the exponent of 1.5 and the constant of 0.4 will both be on the safe side.

Measurements of frictional torque on a well designed instrument indicate that the curve, Fig. 6 in the article, checks well with present prac-

Comparative Index of Radial Type Ball Bearings

ROBERT COREY DEALE
Babylon, N. Y.

● Type numbers used by various ball bearing manufacturers do not always conform to S.A.E. standard designations. This situation often is confusing, particularly since all manufacturers do not offer all types of bearings. The accompanying table lists only those type numbers corresponding to S.A.E. designations. Many additional types are available, as offered by the various ball bearing manufacturers only ten of which are included in this table. The three series numbers for each of the six groups designate bearings for light-duty, medium-duty and heavy-duty service.

Type of Bearing	SAE	MRC Gurney New No.	Gurney Old No.	Strom	New Departure	Hess Bright SKF	Fafnir	Schubert	Federal	CJB	Norman Hoffmann
Single-Row Radial	200 300 400	200-M 300-M 400-M	200 Rad. 300 Rad. 400 Rad.	200-N 300-N 400-N	1200 1300 1400	1200 1300 400	200-W 300-W 400-W	200-N 300-N 400-N	1200-M 1300-M 1400-M	200 300 400	MT-200 MT-300 .....
Single-Row Deep Groove	200 300 400	200-C 300-C 400-C	200-C 300 C 400 C	200 300 400	3200 3300 3400	6200 6300 6400	200 K 300 K 400 K	200 300 400	1200 1300 1400	6200 6300 6400	200 300 400
Single-Row Angular Contact	7200 7300 7400	7200 7300 7400	200 RT 200 300 RT 200 400 RT 200	7200 & 7200-P 7300 & 7300-P 7400 & 7400-P	0200 0300	7200 7300 7400	7200 7300 7400	7200 7300 7400	7200 7300	4200 4300 4400	....
Double-Row Radial	5200 5300 5400	5200-R 5300-R 5400-R	200 DR 300 DR 400 DR	....	....	5200 5300 5400	....	5200 5300 5400	....	5200 5300 5400	....
Double-Row Angular Contact	5200 5300 5400	5200 K 5300 5400	200 DR 200 300 DR 200 400 DR 200	5200-A 5300-A 5400-A	5200 5300	5200 A 5300 A	5200 5300 5400	9200 9300 9400	5200 5300 5400	....	....
Single-Row Narrow Radial	....	100-N 1-N 50-N	....	100-N 1-N 50-N	....	....	100 1 a 50	100-A	....	....	....
Extra Small	8 36	 36	....	 36	ND 8 36	E 8 36	8 36	 36	FB 8 9432	.....	E 8 C-96

Courtesy of the Norton Company, Grinding Machine Division

tice. For a weight of 5 gm. the measured friction was 0.084 dyne-cm., comparing with a value of 0.07 given in the curve. —B. E. LENEHAN
Meter Engineer
Westinghouse Electric & Mfg. Company

To the Editor:

● On page 179 in the article appearing in the May number on jewel bearings, Fig. 5 gives the radius of the pivot in centimeters. In the original curve appearing on page 21 of the *National Physical Laboratory* report, "An Investigation of the Problems Relating to the Use of Pivots and Jewels in Instruments and Meters," the pivot radius is given in millimeters, so that the numerical values for the pivot radius as given in the *Product Engineering* article are ten times too large.
—V. STOTT
National Physical Laboratory, England

Editor's Note—We regret this error in our curve giving the minimum pivot radius for a given pivot load. The pivot radius referred to in Mr. Lenehan's discussion is thus approximately twice rather than $\frac{1}{3}$ of the minimum radius.

Mental Perspective in Problem Analysis

P. H. BRYANT
South Euclid, Ohio

● Recent articles in *Product Engineering* have suggested systems of problem analysis following a fixed routine of thought through to a point where a line can be drawn and the algebraic sum of the separate factors then stand forth as the probable solution of the problem.

Although the value of an orderly procedure in the development of constructive thought cannot be overestimated, too much attention is being centered on details of such systems, and not enough on the mental characteristics essential to the system. The tendency of some of the articles is to create the impression that a given list of questions will apply broadly to all sorts of problems, and when fully answered will cover all of the contingencies. Such an impression is misleading. In the field of creative design, no blanket catechism will serve a useful purpose.

Methods of attack must be suited to each separate individuality. It is only rarely that a designer can use the working formula prescribed by another, since each designer must develop his own method, adapted to his temperament, habits of thought and class of problems.

One essential mental characteristic of any method of problem analysis is the ability to form a mental image in true perspective of the problem even though those essentials may not yet have real form or substance. The designer must have the ability to detach the problem from its environment, to set it off in a corner with a contrasting background. In solving problems of interference, this habit is especially helpful, requiring only directed thoughts.

Many types of problems peculiar to engineering are readily solved in this manner once the practice of mentally giving form and substance to tentative designs becomes a matter of routine.

Pin and Shaft of Equal Strength

To the Editor:

● Although it is interesting, the analysis given on page 187 of the May number in the article by H. F. Bariffi is of little practical value in the design of safety shear pins.

If Equation (5) is solved by Newton's approximation rule, the ratio r_1/r_2 is found to be 0.397. This solution applies to all shafts which meet the assumptions given in the article, and it is apparent that such a pin is so large in proportion to the shaft that the assumptions cannot be considered safe. With such a large pin the shaft section is far from circular, and the torsional strength cannot be determined by Equation (2) as assumed.

Whenever a pin of reasonable diameter is used in a shaft of the same or stronger material, it is apparent from the ratio of r_1 *to* r_2 that the shaft will stand a load greater than the shearing strength of the pin.
—NORMAN M. WICKSTRAND
Meriden, Conn.

Market Investigation by Questionnaire

To the Editor:

● General Motors engineers may not follow all the ideas indorsed by the majority of replies to their questionnaire but if they had relied merely upon their own judgment, many popular design changes might have been overlooked. Even though their executives think they know what is best for the public, no company can afford to ignore the opinions of their prospective customers.

Every member of a concern engaged in the development of a product holds a biased opinion, whether he wants to or not. Furthermore, the public acceptance is variable. Today an article may be acceptable while a few weeks earlier or later it may be rejected. Likewise, an idea may "take" in one locality and fail in another.

All concerns cannot use the same manner of approach as the one General Motors used, but there are many other ways of finding out these customer preferences. At least, the field for a new product should be carefully investigated before investing in tools and equipment. Otherwise, conditions may develop as in the example of a concern that produced a small machine for use in a specific field. Proceeding on the assumption that the machine was useful and practical and that it could be sold at a certain price, an expensive tooling program was carried through to obtain this low price. After the machines were produced it developed that less than half of the machines could be sold in a reasonable time and at a normal expense. Investigations then disclosed a limited market at the established selling price. This condition could have been determined beforehand by the use of a market investigation using a questionnaire similar to the one described on page 21 of the January number of *Product Engineering*.
—M. G. DEMOUGEOT
Assistant Factory Manager
International Banding Machine Co.

Fillet Profiles

To the Editor:

● Optical difficulties in making accurate measurements of stresses at boundaries limit the accuracy with which boundary stresses can be obtained, according to the article, "Stress Concentration Produced by Holes and Notches," appearing on page 92 of the March number. Dr. R. V. Baud in his article in the April number on "Fillet Profiles for Constant Stress," does not state what photo-elastic material, or what method of measuring stress was used, or within what limits of error constant boundary stresses were obtained. Unless elaborate precautions were taken, the accuracy of these stress determinations might not be within plus or minus 5 to 10 per cent.

The use of the phrase "hydrodynamical analogy" in the Baud article seems unfortunate, as the word "analogy" has ordinarily been limited to applications in which an exact mathematical relation has been shown to exist. If such an exact analogy were known to exist in the case under discussion, photo-electric tests would be superfluous. In reality, the author uses a practical "cut-and-try" method,

and takes the contour of a free jet of fluid as a likely starting point. His treatment of the bending problem is explicitly the cut-and-try method.

—E. E. WEIBEL
University of Michigan

Shaft Deflections

To the Editor:

● In his article, "Determining the Critical Speeds of Shafting," in the June number, W. C. Willard uses the term "maximum deflection" in calculating the critical speed, whereas his Equation (1) is derived on the basis of the deflection under the load. The maximum deflection of the shaft does not always occur under the load.

—E. S. AULT
Case School of Applied Science

To the Editor:

● The term "total deflection" rather than "maximum deflection" might have been clearer. The shaft deflection at the load caused by all loads on the shaft was the idea that the term was intended to convey.

As published, the article omitted units assigned to constants in Equations (2), (3) and (5). The weight of the concentrated load W, and the weight of the shaft S are both expressed in units of 100 lb. In Equations (7) to (14) inclusive, the published article also omitted the units assigned to R. Changing the constant 4.263 to 4,263, the reaction R is in terms of "kips," or units of 1,000 lb. As published, the values of R are in units of 1,000,000 lb., a unit inconveniently large. —W. C. WILLARD
Buffalo Forge Company

Natural Logarithms

To the Editor:

● In the article, "Fillet Profiles for Constant Stress," appearing on page 133 of the April number, the last equation given on the page does not check with the values given in Table II in the article. Is the formula correct as shown?

—CARL P. NACHOD
Vice-President
Nachod & United States Signal Co.

Editor's Note—The equation referred to by Mr. Nachod involves natural logarithms, and therefore the term *log* in the equation should carry the subscript *e*. The values in the table are properly calculated on the basis of natural logarithms. In scientific work the natural log is more commonly used, while in mechanical design the base 10 is ordinarily involved. In order to avoid this misunderstanding, we will hereafter indicate the base of the logarithm.

Velocity Factors for Various Types of Gears

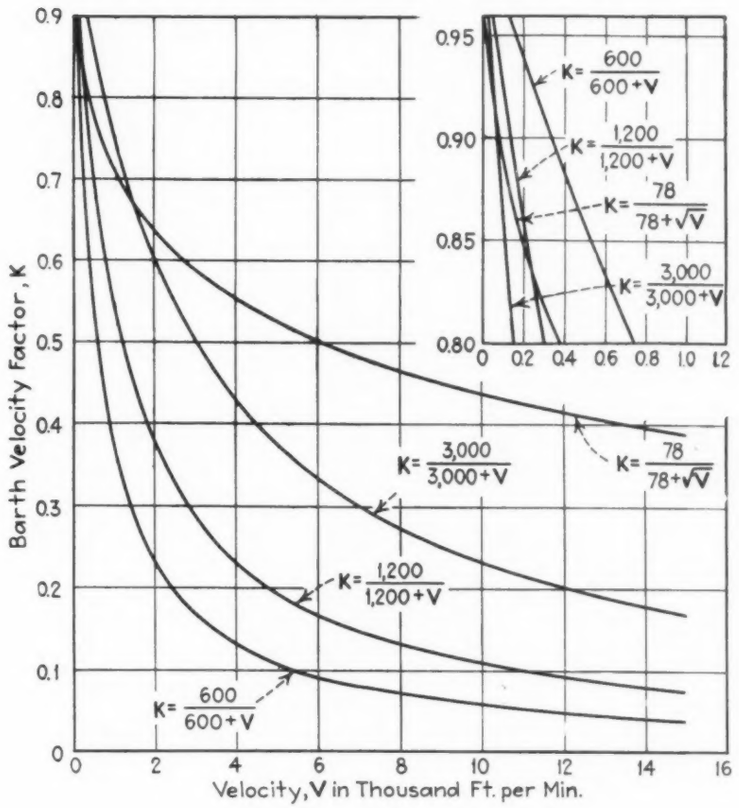


Chart for Barth Gear Factors

FRANK A. MICKLE
Associate Professor, University of Michigan

● Data given in the tables of Barth Gear Factors on page 160 of the April number can be presented in a more convenient form on charts. Since the basic value is velocity and the secondary value is the velocity coefficient, these factors form the coordinates for the curve reproduced in the accompanying chart.

Most commonly used for straight spur and bevel gears, the equation with the factor 600 gives the lowest velocity coefficient. For helical gears and ground straight-tooth gears, the curve based on 1,200 is also shown, along with a similar curve for helical and herring-bone gears, in which the factor of 3,000 is used. However, in the *A.G.M.A. Recommended Practice for Herringbone Gears,* this last relation has been replaced by the one in which the factor 78 appears.

The upper right-hand portion of the chart is a continuation of the main chart with the velocity factors drawn to an extended scale.

Names for Materials

● Quoting from one of many letters received, "Referring to your editorial 'A Job for Engineering Societies to Start,' we believe that a nomenclature of materials based on applications would do away with much annoyance and unnecessary arguments. All metallurgists and engineers will undoubtedly be heartily in favor of this move proposed by *Product Engineering.*"

Undoubtedly many more of our readers have definite thoughts on this subject. The question is not whether trade names should be abolished. It relates to the feasibility of establishing an intelligent nomenclature for materials.

In a forthcoming number we will publish the opinions on this question as expressed by our readers and representatives of engineering societies.

Hinges and Pivots for

HENRY J. MARTIN

Fig. 31—Common steel hinge applied to a machine tool cover.

Fig. 32—Cover lugs cast integrally and pivot pins fastened in loose piece for greater span.

Fig. 33—When swinging member *A* must be removable, bearing is cut as in upper view.

Fig. 34—Swinging rod retained by pinned collar *A*. Both lugs are integral with the casting.

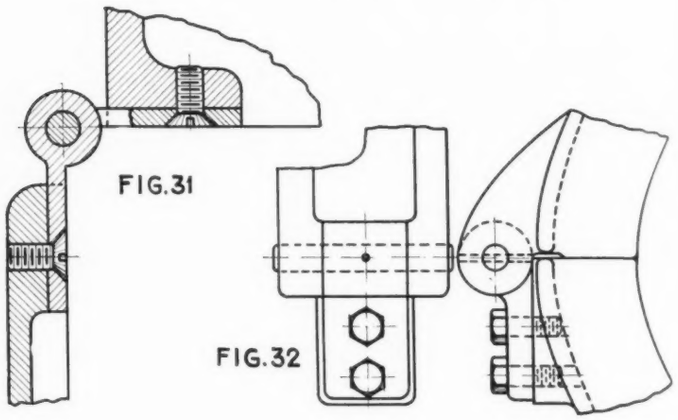
Fig. 35—Pivot bearing as used on an adjustable vise jaw.

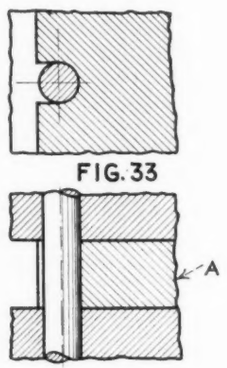
Fig. 36—Toggle or pawl joint. Angle *A* should be 30 to 45 deg. to retain the member.

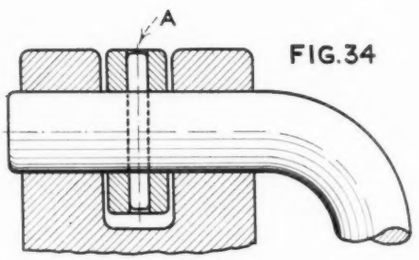
Fig. 37—Radial and axial play are taken up by the hardened and ground bushings tapered to an included angle of 22½ deg., sufficient to prevent sticking.

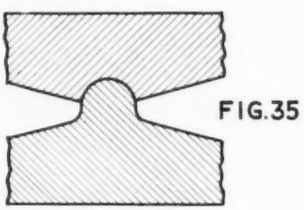
Fig. 38—Needle bearing pivot for either rotation or oscillation, with three hardened and ground washers for separating the rollers.

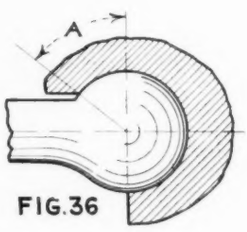
Fig. 39—Three ball joints used in a gear shift mechanism. Hole *A* is in shifting rod, *B* is the pivoting center which is retained by the inserted locating plug at *C*.

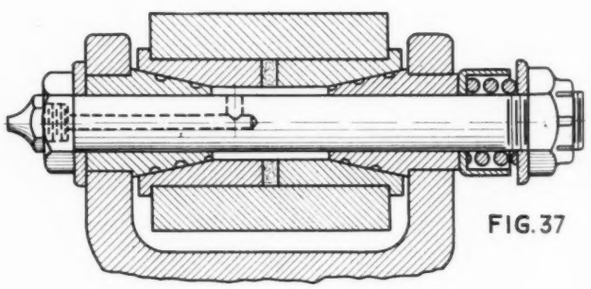


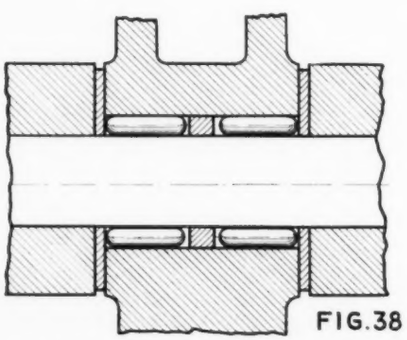


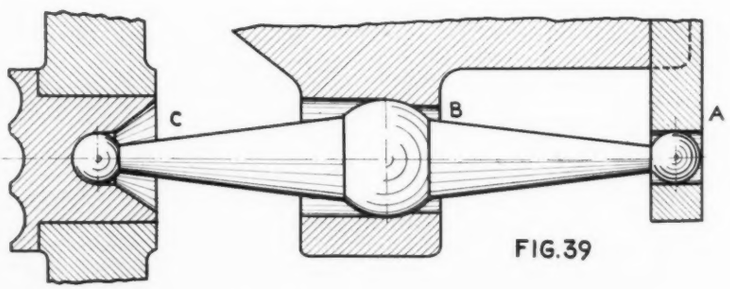












Covers and Flexible Joints—II

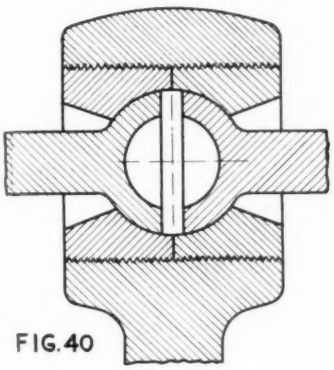


FIG.40

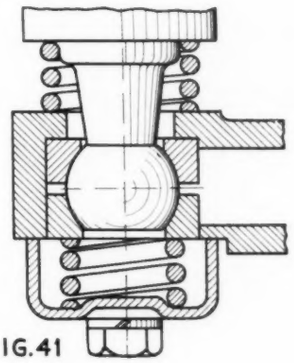


FIG.41

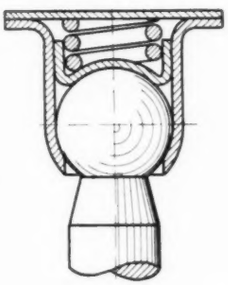


FIG.42

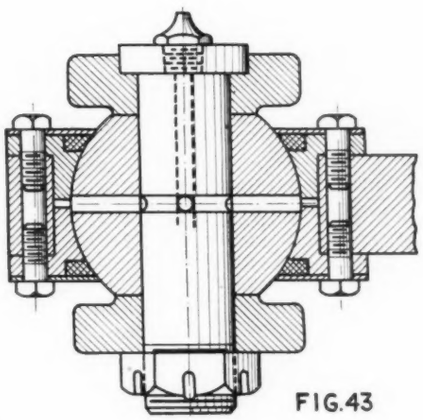


FIG.43

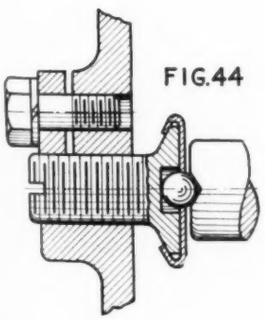


FIG.44

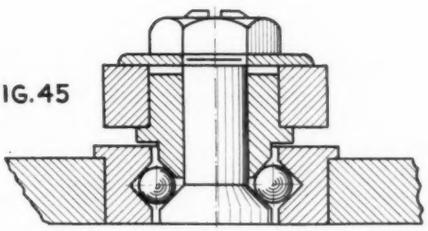


FIG.45

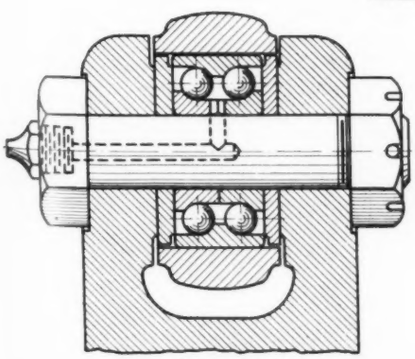


FIG.46

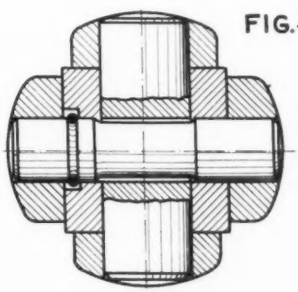


FIG.47

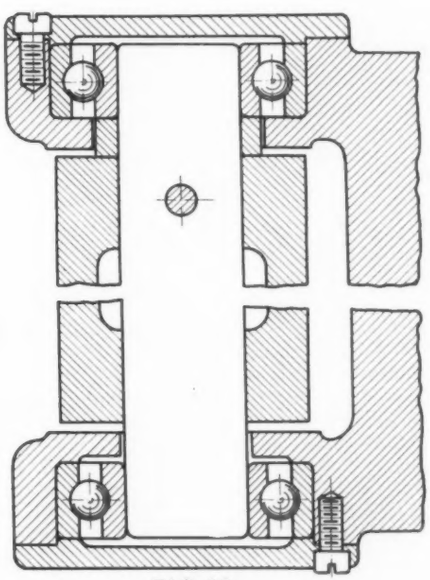


FIG.48

Fig. 40—Socket joint with hemispherical rod ends held in place by screw bushings.

Fig. 41—Self-adjusting socket joint. The sheet metal spring cover is held in place by two screws.

Fig. 42—Sheet metal ball socket housing with cover fastened by spot welding.

Fig. 43—The flattened sphere is held by the center stud. Felt seals are used to retain the grease.

Fig. 44—Combination pivot joint and end thrust bearing, ball being retained by the washer spun over the fixed screw.

Fig. 45—Arm joint of a pantograph machine, with center stud clamped without end play, stud head and bushing end forming the ball race.

Fig. 46—Rocker arm bearing as used on airplane.

Fig. 47—Universal joint, the smaller pin being retained by wire snap ring.

Fig. 48—Arm joint designed for accuracy. Upper ball bearing takes all thrust caused by weight and the spindle is pinned to the stationary member. The bearing has a light press fit.

NEWS

Shortage of Trained Engineers Is Predicted

Technical colleges graduating fewer men while scope of engineering field is continually broadening

IN ANSWER to the question, "Is engineering overcrowded?" Dr. Harvey N. Davis, president of Stevens Institute of Technology, emphatically said "No," and backed up his answer with much convincing data and logic. His address was the first of a series of talks by college presidents and deans, ushering in the Choosing-A-Career Conference held at L. Bamberger & Company, in Newark, N. J., and was broadcast over Station WOR on May 31. Dr. Davis emphasized that one must not let the present temporary condition warp one's judgment as to permanent trends. The real question is, "Over the working lifetime of a young man now choosing a career, will engineering be overcrowded?"

In his answer to this question, Dr. Davis referred to the study made last winter by the statistical staff of the American Telephone & Telegraph Company, under the direction of General Rees, vice-president of that company. A curve was constructed showing how many engineers are being graduated, year by year, from all of the schools of the country over the period from 1920, through 1937. The last four years of this curve are based upon the number of men entering colleges, corrected for the probable number of non-graduates as indicated by past records. From slightly less than 9,000 graduates in 1921, the trend of the curve is upward, indicating 11,000 graduates in 1937. From 1927 to about 1933 the number of engineers graduated annually was above the trend line, while for 1934 the total number graduating from engineering schools will be definitely below the trend line. By 1937 there will be graduated something like 3,500 fewer men than indicated by the normal trend line.

Referring to a remark made about a year ago by Dr. James A. Farrell, former president of the United States Steel Corporation, to the effect that American industry would soon find itself facing the worst shortage of technically trained men it had ever known, Dr. Davis stated that already occasional straws are showing which way the wind is blowing. As an example, nearly three-quarters of last year's graduating class at Stevens now have jobs, and the Alumni secretary at Stevens recently stated that he is suddenly beginning to have real trouble in finding enough keen young Stevens graduates with four or five years of actual experience to match the opportunities that are coming to his desk.

Another reason advanced by Dr. Davis for believing that engineering is not going to be overcrowded during the coming quarter century is that the scope and range of the engineering field is continually broadening. Not only will there be a considerable reabsorption of men as a result of the developments taking place in all fields of industry, but management and industrial engineering, historically children of mechanical engineering, are now beginning to draw heavily upon trained engineers for their personnel.

In closing his address, Dr. Davis said, "Four years from now there will not be enough engineering graduates to meet even the technical demands of the profession, to say nothing of all the other opportunities for engineers."

Temperatures Affect Wear Rate

HARDENED and tempered plain carbon steels, tested in a hydrogen atmosphere on the Amsler wear test machine, show an abrupt transition from one rate of wear for specimens tempered at a low temperature to an extremely high rate of wear for specimens tempered at the higher temperature. According to tests conducted by the National Bureau of Standards, the wearing of specimens tempered at the higher temperatures results in a bright, rough surface and the rate of wear is extremely high.

It is indicated by these tests that the tempering temperature which corresponds with the change in type of wear increases with increasing carbon content of the steel. For 0.4 per cent carbon steel, the transition temperature is close to 200 deg. C; for a 0.8 per cent carbon steel, it is at about 300 deg. C., and is between 400 deg. and 500 deg. C., for a 1.3 per cent C steel.

These studies have been made, for the most part, under a given set of conditions with reference to the contact pressures between the specimens and the amount of rubbing between the two specimens rotating at slightly different speeds.

New Effects with Steel-Faced Plywood

GREAT crowds are being attracted to the new Ford Building at the Century of Progress. Many people consider it to be the most attractive and magnificent structure at the Fair. Such an impression is largely the result of striking effects created through the use of a new type of architectural treatment for columns, entrances, interior panels and trim.

Large areas of wall surface are covered by pre-fabricated plywood sheets, $\frac{3}{8}$ in. thick with a 14-oz. alloy-steel face and a galvanized-steel back. Decorative bands around the interior court have a serrated or corrugated face providing an unusual effect. Entrance and court columns are formed of sheets.

Kunial Alloys

FURTHER information on the recently announced Kunial alloy developed in England discloses that this brass in an age-hardening form is a material of a type already well known among American metallurgists. In fact, the physical properties claimed for the Kunial alloy hardened by heat-treatment from either the soft or cold-worked condition are hardly equal to the properties found in commercially available nickel-silicon-copper and beryllium-copper alloys.

According to W. H. Bassett, metal-

lurgical manager of the American Brass Company, "The Kunial material is an alloy of copper, nickel and aluminum in which the aluminum is generally not more than one-fourth the nickel content. The American equivalent to the Kunial alloy is Tempalloy, a product of the American Brass Company, in which silicon and nickel are used for precipitation hardening, the silicon usually being about one-fourth the nickel content. In Kunial it is usually considered that hardening is accomplished with Ni_2Al; with Tempalloy the hardening is produced by Ni_2Si, although it is customary in Tempalloy to add a small amount of aluminum in order to overcome scaling in heat-treatment. Either nickel and silicon or nickel and aluminum may be added to copper to bring about hardening, or these alloy additions may be used in connection with such nickel alloys as German silver, nickel silver, as well as with brasses."

As to the future, Mr. Bassett states: "In our opinion we have only begun to hear of alloys of this type, the probability being that both silicon and aluminum will be used in connection with nickel. As the constituents for precipitation hardening, these alloys are annealed or softened by heating to about 850 deg. C. (1560 deg. F.) and quenched. These materials are then hardened by heat-treating at about 450 deg. C. (850 deg. F.). The material may be cold worked when in the annealed condition, heat-treatment at 400 deg. C. (750 deg. F.) producing even greater hardness and strength."

A.S.T.M. Revised Standards

SPECIFICATIONS and tentative standards for cold-rolled strip steel, stainless, magnesium alloys, and welded steel pipe are among those revised at the annual meeting of the American Society for Testing Materials. To agree with N.R.A. Code tolerances, gage tolerances for strip and sheet steel were revised. Recognizing that physical properties alone do not indicate the drawing properties of sheet steel, this part of the revised standard is now included in an appendix.

Tentative standards on welded corrosion-resistance articles of austenitic stainless steels now specify heat-treatment at not less than 1,800 deg. F. for a period of at least one hour for each ½ in. in thickness, followed by rapid cooling in air. This treatment not only increases the resistance to corrosion, but also relieves internal stresses developed during manufacture. The standard specifies that no further welding or heating shall be performed subsequent to the annealing operation.

Alloy-steel nuts for high-temperature service, are to be cold-cut or hot-forged and cold-trimmed; nut cuts from drawn or rolled bar stock are not permitted under new revisions of this standard. Tentative specifications were also proposed for electric-fusion-welded steel pipe, for high-temperature and high-pressure service. This specification includes physical tests, radiographic examination, and stress-relieving treatment.

Revisions were also presented for tentative specifications of magnesium alloys, covering eleven compositions. Five of these alloys are for castings, two are for rolled sheet material, while five are suitable for forging.

Carnegie Tech to Teach Industrial Art

RECOGNIZING the need in industry for artists trained to design products of utility to be made in quantity production, Carnegie Institute of Technology is introducing a course of industrial design in the Department of Painting and Design. According to Prof. W. A. Readio, chairman of the Department of Painting and Design, progressive manufacturers have shown a trend, especially marked in the last five years, toward a consideration of artistic quality on an equal footing with mechanical improvements. The artist's contribution to the motor cause, glass, utensil and appliance fields has established him as indispensable in the creation of modern quality products.

According to Professor Readio, the traditional view that the artist is a handicraftsman whose existence is threatened by the debasing machine is a misconception which has often barred the artist from a field logically his. It is the primary aim of the new course to develop a type of artist-designer who will look upon the machine as a means for making useful articles possessing refinement of form and attractive appearance.

Included in this course is a thorough study of the principles of design, drawing and the history of the arts and crafts up to modern times. With this background, the students will spend the latter half of his four years studying artistic expression as modified by the nature of quantity production, the character and limitations of the various materials, consumers needs and marketing problems. It is expected that the student so trained should be prepared to meet the needs of a particular industry with a minimum of time lost in training at the plant.

MEETINGS

National Metal Congress—New York, N. Y., Oct. 1-5, the following societies participating:

American Society for Metals—Hotel Pennsylvania, October 1-5. W. H. Eisenman, secretary, 7016 Euclid Ave., Cleveland, Ohio.

American Welding Society—Hotel New Yorker, Oct. 1-5. Miss M. M. Kelly, secretary, 29 West 39th St., New York, N. Y.

American Institute of Mining and Metallurgical Engineers—Institute of Metals Division, Hotel Pennsylvania, Oct. 2-4. John T. Breunich, assistant secretary, 29 West 39th St., New York, N. Y.

EXHIBITIONS

National Metal Exposition—Port Authority Bldg., New York, N. Y., Oct. 1-5. W. H. Eisenman, director, 7016 Euclid Ave., Cleveland, Ohio.

Power Show—11th National Exposition of Power and Mechanical Engineering, Grand Central Palace, New York, N. Y., Dec. 3-8. Charles F. Roth, director, Grand Central Palace, New York, N. Y.

New System of Speed Ratios

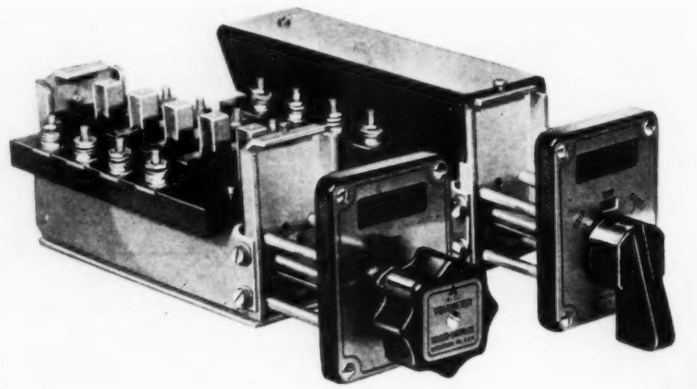
BASED on a 22½ per cent increase from one ratio to the next higher ratio, instead of the 25 per cent obtained with the use of "10" series, a new system of standard electric motor speeds was suggested by B. S. Weaver, before a recent meeting of the American Gear Manufacturers Association. The new series is said to permit the use of the same development for 60, 50, 40, 30 and 25-cycle motors, with synchronous speeds of 1,800, 1,500, 1,200, 1,000 and 800 r.p.m., with a deviation of approximately 2 per cent in the output speeds of the 50 and 25 cycle motors.

This system of ratios will require the "40" series. The disadvantage in the use of the "10" series arises principally in applying motors where odd frequencies are encountered, making it necessary to increase the number of ratios to give low speeds reasonably approaching the speed obtained with 60-cycle motors.

New Materials and Parts

Roller-Smith Control Switches

Designated as Type R, for use on switchboards and control panels, for connecting electrical measuring instruments and for operating control switches. Current carrying contacts are rated at 15 amp. 150 volts, a.c., and are insulated for 600 volts. Self-aligning moving contact bridges two fixed contacts making pigtail and spring-type connections unnecessary. Contacts are silver-faced and mounted in a molded Bakelite structure supported on two formed steel side-plates, held between steel end plates which carry bearings for the rotating shaft. Bakelite cam mounted on the shaft engages an indexing plunger. Available with either round, notched handles or with finger-grip handles, with a mechanical indicator showing a red or green marker to indicate the last operation. Designed for mounting on panels from $\frac{1}{8}$ to 2 in. thick. Body of switch is 3x3$\frac{5}{8}$ in. Length back of panel varies from 6$\frac{3}{4}$ in. for 2 contacts, to 10$\frac{7}{8}$ in. for 8 contacts. Supplied in various combinations of single and double poles. Roller-Smith Co., 233 Broadway, New York, N. Y.



Limitorque Valve Operator

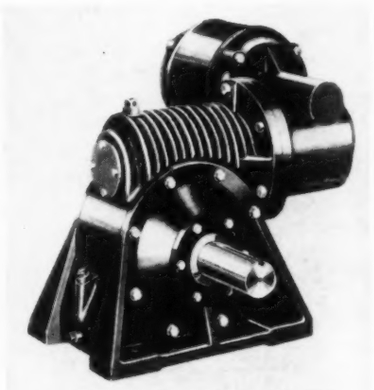
Consists of a self-contained electric motor, gear reducer and limit switches. Valve operating nut raising and lowering the valve disk is carried on tapered roller bearings and driven through a worm shaft geared to an electric motor through a helical gear drive. Totally inclosed motor operating at 1,800 r.p.m. is flange-mounted to the gear case. Gear-driven mechanical limit switch stops the motor when valve reaches the full open position. Operation in the closing direction is limited by a torque limit switch. The worm is driven by splines on the worm shaft and end-wise movement is normally opposed by a heavy coiled spring. When the valve closes or when it meets an obstruction at any point in its closing motion, end thrust on the worm compresses the helical spring, and a flange on the worm opens a snap switch. Available for valves of 2 to 72 in. in diameter, and for pressures up to 1,000 lb. per sq.in. Said to be adaptable to every form of gate and globe valve, door operator, sash operation and for other mechanism having a definite movement that must be stopped or reversed at the end of travel. Philadelphia Gear Works, 330 West 42d St., New York, N. Y.

Tubular Lamps

Designed to be placed end-to-end to form a continuous line of light for architectural and fixture applications, the lamps incorporate a new type of seal between the glass tube and metal base, eliminating basing cement. At each end of the tubular lamp a chrome iron contact cap is sealed directly to the glass tubing. Within the tubing a single stretched-coil filament is drawn into a continuous line from one end-contact cap to the other, held by wire filament supports attached to a channel backbone inside the bulb. Available in 30- and 60-watt units, with an overall length of 17$\frac{7}{8}$ in. and a diameter of 1 in. Designed for a life of 1,500 hr. General Electric Co., Nela Park, Cleveland, Ohio.

Janette Speed Reducers

Motorized double worm-gear reduction units available with motors up to 1 hp., having ratios ranging from 96 to 1, up to 8,100 to 1. Supplied with direct current, single-phase or polyphase motors. All gear shafts operate on tapered-roller bearings. Motors are ball bearing. Polished steel worms and bronze gears designed for high-torque applications. Janette Manufacturing Co., 556 W. Monroe St., Chicago, Ill.



Porcelain Indicating Cutouts

Plug and cartridge fuse bases have integral neon lamp indicators. When the fuse blows, neon lamp glows, indicating fuse to be replaced. Inserting a new fuse stops the neon glow. Since lamp glows only when the fuse is blown, lamp is said to have long life. Lamp draws no measurable current,

and indicates open fuse with or without load on the circuit. Mounting holes and wiring terminals are interchangeable with standard types and available for 30, 60 and 100 amp. capacities. Finished in green porcelain. L. S. Brach Mfg. Corp., Newark, N. J.

G-M Weatherproof Photoelectric Relay

Designed for outdoor use, relay is completely inclosed in a cast aluminum case, with a hinged cover sealed with a gasket. As illustrated, cover can

be fitted with tubular visors to shield out undesirable light and to protect the windows from the weather. Light from a lamp behind the upper visor is focused on the weather-protected mirror and reflected back to a photoelectric cell behind a lens in the lower visor. Relay designed for operation on 110 volt, 60 cycle current only. Relay contacts are rated at 2 amp. a.c., non-inductive loads; speed of operation 600 per min. Photoelectric cell is of the caesium type, gas-filled. Case dimensions are 13 in. high, 10½ in. wide, 6 in. deep, not including visors. G-M Laboratories, Inc., 1731 Belmont Ave., Chicago, Ill.

Magnolia Bronze Bushings

Semi-finished outside and inside, in standard lengths of 12 in., bearing bronze bushings are available in a material corresponding to S.A.E. specification No. 64. Physical properties are: Brinell 55, tensile strength 25,000 lb. per sq.in., elongation in 2 in. 8 per cent. Composition is said to provide excellent bearing qualities combined with strength and toughness. Available in solid and cored bars from 1 to 5 in. outside diameter. Outside diameter is 1/16 in. oversize, inside diameter 1/16 in. undersize. Also available in 13 or 14 in. lengths. Magnolia Metal Co., 120 Bayway, Elizabeth, N. J.

Ideal Sealing Compound

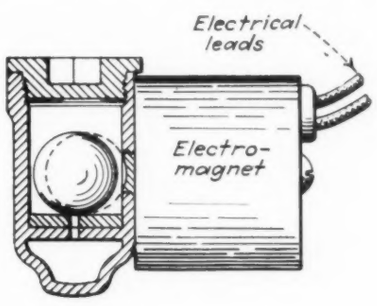
For use with or without a gasket to seal metal joints against pressure or vacuum, and as a lubricant for threaded joints. Compound has a castor oil base, free of grit. Said to be non-soluble, non-corrosive and non-oxidizing, and to resist temperatures up to 400 deg. F. Also said to protect electrical connections and battery terminals from chemical corrosion. Recommended by the manufacturer for use in fabrication or assembly of mechanical units that are required to be easily and quickly disassembled. Available in 3 and 8 oz. tubes and in drums up to 135 lb. Ideal Commutator Dresser Co., Sycamore, Ill.

Mercury Tube Switch

Plunger extending through the side of the case is arranged to tilt the mercury tube to make or break an electrical circuit. Designed for use as a switch in the pilot circuit of motor controllers; can be used as a limit switch on motor driven machines. Case is approximately 5x3x1⅛ in.; available in two capacities, 10 amp. 125 volts, and 30 amp. 125 volts, a.c. or d.c. Also available without plunger, operated by rocking the switch. Hart Mfg. Co., Hartford, Conn.

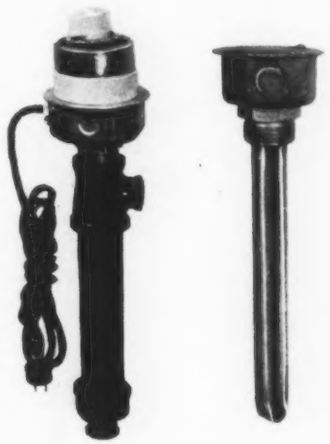
Electromagnetic Valves

For remote control of oil, gas, air, refrigerant and water lines, electrically operated valve is supplied in a bronze



or non-magnetic iron body having a housing on the side inclosing an electromagnet. When energized, the magnet pulls a steel ball horizontally from the valve seat, holding it against a metal wall between the magnet and valve body. The ball, which is the only moving part, is totally inclosed, and there are no packings or plungers. Valve is said to be suitable for use underground or submerged and in explosive liquids or gases. Available in six sizes for pressures from 6 to 225 lb. with all electrical parts sealed within the valves. A. F. Hoppe Engineering Co., 246 S. Meridian St., Indianapolis, Ind.

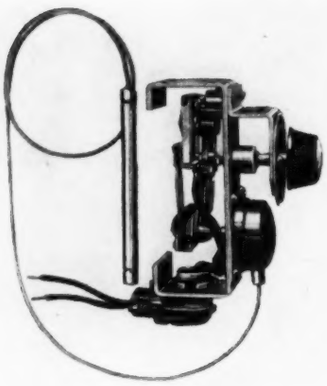
Trent Immersion Heaters



Supplied with a body of seamless brass or copper fabricated from one length of tubing and welded into the head, as shown at the right in the illustration. Outlet box attached to the head is removable and can be rotated to suit the position of the wiring. Thread under the head is 1¼ in. standard pipe thread. Heater can be connected for one or three heats, the three-heat unit also supplied with a switch attached to the outlet box. Bodies can be supplied lead plated or in steel or chrome alloy as required, to resist corrosion. Available in body length from 6 in. to 16 in.; in ratings from 500 to 2,500 watts, 115 or 230 volts. As shown at the left, also available as a side-arm heater with 3-heat switch, cord and plug. This unit supplied in ratings from 1,000 to 2,000 watts, arranged for ¾-in. pipe connection. Harold E. Trent Co., 640 North 54th St., Philadelphia, Pa.

Cabinet Thermostat

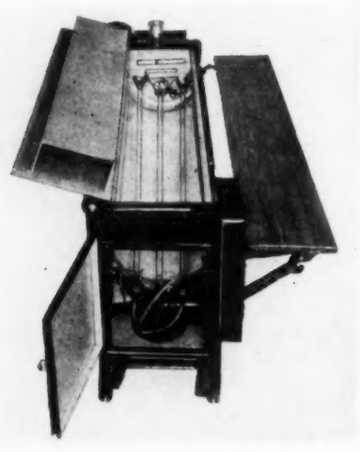
For inclosed type heating, cooling and air conditioning units, thermostat is arranged to break an electric circuit on increase of room temperature to control an electric heating element, a gas valve or fan. For cooling units, the thermostat is arranged to make the circuit on increasing temperature. Thermostat is installed completely inside cabinet except for the exposed dial and knob adjustment. Relay is of the high-voltage type, capable of controlling motors up to 1 hp., single phase, 110 and 220 volts, a.c.; or ½ hp.



on 110 and 220 volts, d.c. Bakelite knob completely covers the mounting screws. Assembly may be mounted on vertical or horizontal surface. Available as standard with adjustable ranges for heating from 45 to 80 deg. F.; for cooling from 75 to 95 deg. F. Made only in single pole, single throw type, either making or breaking circuit on increase of temperature. Detroit Lubricator Co., Marquette & Viaduct Sts., Detroit, Mich.

Blue Print Machine

Tracings up to 42 in. in width and to any length can be printed with the Model E Blue Print machine. A half cylinder of curved plate glass is bedded into the end frame castings and around this glass travels an endless canvas band. Continuous travel of the band brings the sensitized paper and tracings within the printing range of the three General Electric vapor tube lamps equipped with an aluminum satin finished reflector. A $\frac{1}{6}$ hp. motor drives the band and a variable speed control gives the speed variations in 6 in. per min. to 5 ft. per min. After passing around the half cylinder of glass, the tracings are returned to the operator. Blue prints, Van Dyke negatives, blue line prints, black and white prints, or Ozalid prints can be made with this machine. Total power consumption is 1,768 watts for the three lamps and the motor. Shaw Blue Print Machine Co., Inc., 9 Campbell St., Newark, N. J.



Jewel Repeating Interval Timer

Designed for use on electrical equipment requiring continuous interval operation. Timing mechanism is driven by a small, non-synchronous 110 volt, a.c. motor, through a train of gears. Two operating cams are mounted on a shaft rotating one revolution in 30 min. The rear cam is permanently attached while the other cam is adjustable in relation to the fixed cam. Circuit-closing periods

from 1 sec. to 8 min., over the full cycle of 30 min. may be obtained. Cams operate quick-break single-pole switches with silver contacts. As illustrated, the timer is mounted on a cast-iron base 4x4 in., with screw holes in corners for mounting. Grooved surface in the base is fitted with a rubber gasket sealing a sheet metal cover to make a dust-proof housing $2\frac{1}{2}$ in. high. Contacts rated at 3 amp. Designated as type 10-360-8. Walser Automatic Timer Co., Chrysler Bldg., New York, N. Y.

Hose Joints

Substituting nipples and flanges, hose joint eliminates all contact between metal and fluid, permitting greater flexibility. It affords a perfect seal in all suction or discharge service up to 125 lb. working pressure and is recommended for the larger diameters of suction or discharge hose handling abrasive materials or corrosive liquids. The end built into the hose consists of an enlargement, or head, reinforced with numerous plies

of fabric surrounding a rigid steel ring of angular cross section. The joint is assembled with the aid of split flanges and standard bolts. Two successive lengths of hose can thus be bolted together with the rubber ends of the hose compressed to form a seal. Spacing of the bolt holes is standard, so that the end of the hose can be fitted to the end of a standard pipe. When the bolts are drawn up the pressure of the split flange against the flared hose end provides a positive and unyielding clamping of the joint. Joints are furnished in sizes $1\frac{1}{2}$ in. and up. The B. F. Goodrich Rubber Co., Akron, Ohio.

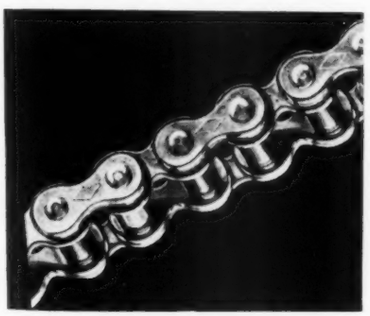
Vertical Motoreducers

Embodying a special design using a self-priming, reversible oil pump, with seals and oil throwers designed to prevent escape of lubricant. For double or triple reduction, one motor end-bell is removed and the frame coupled through an adapter to the gear case, forming an integral unit. Gears are of precision, single helical type, of alloy steel. Ball or roller bearings are used, with low speed shaft bearings designed to handle overhung loads. Sleeve

carrying the output shaft bearing extends above the oil line, the bearing being separately lubricated. Oil is circulated by two sets of helical gear rotors arranged so that one or the other is effective, depending upon the direction of rotation. Pump is self-priming, driven from the first reduction gear, pumping a given quantity of oil regardless of viscosity. Oil is distributed through spray piping to gears and bearings, on the incoming side of the mesh point. Oiling system is said to reduce churning losses, giving an efficiency of 97 per cent or over. Available in double reduction ratios from 9.7 to 41.9, and from 45 to 288 in triple reduction. Supplied with motors from ¾ to 75 hp.; 1,750 r.p.m. standard with other speeds available. Supplied only for downward shaft extension. The Falk Corp., Milwaukee, Wis.

Stainless Roller Chains

Roller chains made of 18-8 stainless steel throughout in standard sizes. Advantages claimed for this roller chain are appearance, sanitation and service life. They resist the corrosive action of fruit and vegetable juices, do not



discolor even in food industry applications where the chain must pass through the processed food, and they are easier to keep clean. Diamond Chain & Mfg. Co., Indianapolis, Ind.

Nickel-Aluminum Bronze

A nickel-aluminum bronze whose composition is approximately 92 per cent copper, 4 per cent nickel, and 4 per cent aluminum. Claimed to have unusual corrosion-resistance. Can be supplied in tubing, pipe, sheet, rod and some wire sizes, has good cold-working properties and is ductile and malleable. Rods can be drawn to a tensile strength of about 100,000 lb. per sq.in., and yet remain ductile enough to be cold headed and roll threaded for manufacturing bolts. The hardness, when measured on the Rockwell F scale, varies between 70 and 110, depending on temper. Tensile strength ranges from 50,000 lb. per sq.in., in the soft annealed condition to 115,000 lb. per sq.in., for hard rolled sheet and about 150,000 lb. per sq.in., for very hard drawn wire. The elongation varies from 70 per cent in two inches to 12 per cent for drawn rods and tubes for tensile strength ranging from 50,000 to 120,000 lb. per sq.in. Chase Brass & Copper Co., Waterbury, Conn.

MANUFACTURERS' PUBLICATIONS

Bakelite — Bakelite Corp., Bound Brook, N. J. Small booklet, 24 pages, recounting the development of Bakelite synthetic resins for paints and varnishes, giving information on resistance to corrosion and abrasion, and data on applications.

Blower Wheels — B. F. Sturtevant Co., In., Hyde Park, Boston, Mass. Folder, 4 pages, giving applications and performance data on blower wheels.

Couplings — The Poole Foundry & Machine Co., Baltimore, Md. Leaflet, 4 pages, describing improved gear-type flexible couplings.

Electrical Control Equipment — Square D Co., Detroit, Mich. New catalog pages for loose-leaf catalog No. 37, include prices and descriptions of industrial switches, panel boards, industrial circuit breaker and switchboard.

Electric Motors — Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. The following five leaflets give application, operating characteristics and construction: "Type CR Repulsion-Start Induction Motors," "Type FR Repulsion-Start Induction Motors," "Type FS Polyphase Squirrel-Cage Motors," "Type FK Direct-Current Motors," and "Type CS Explosion-Resisting Fan-Cooled Squirrel-Cage Motors."

Electric Motors — Allis-Chalmers Mfg. Co., Milwaukee, Wis. Circular, 4 pages, illustrating features and construction of squirrel-cage induction motors.

Electric Motors—The Lincoln Electric Co., Cleveland, Ohio. Circular, 4 pages, giving construction details and outline dimensions for self-protected motors.

Electric Motors—The Master Electric Co., Dayton, Ohio. Leaflet, 8 pages, illustrating a complete line of motors, geared-head motors and generators.

Finishes—Copper & Brass Research Association, 25 Broadway, New York, N. Y. Booklet 15-B, 8 pages, 8½x11, "Coloring Copper, Brass and Bronze," outlining newly developed processes for attaining natural patina on copper as well as data for obtaining other color effects on copper and copper alloys.

Frequency Converters—The Louis Allis Co., Milwaukee, Wis. Booklet No. 509-A, 4 pages, with rating, details of construction and data on selection of motor-driven generators.

Oil Circuit Breakers—Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. Catalog 33, 8½x11 in., 72 pages, "Westinghouse Small Oil Circuit Breakers," includes several pages on general information, connection diagrams, and ratings followed by catalog information, applications, constructional details, and operation of indoor, outdoor and subway oil circuit breakers.

Projection Apparatus — The R. Y. Ferner Co., 930 Investment Bldg., Washington, D. C. Catalog No. 577, 8 pages, describing desk type projection apparatus for inspecting, measuring and photographing profiles or surfaces.

Relays—Ward Leonard Electric Co., Mount Vernon, N. Y. Bulletin 131, four pages, giving features, dimensions and wiring data for heavy duty relays.

Roller Bearings—The Bantam Ball Bearing Co., South Bend, Ind. Catalog, 8 pages, 8½x11, describing several types of roller bearings designed for printing presses. Illustrated is a square box bearing designed to replace bronze boxes without changing the frame.

Stainless Steel—National Tube Co., Pittsburgh, Pa. Bulletin No. 24, 8½x11, 16 pages, "Stainless Steels in the Dairy Industry," gives the properties of 18-8 alloy steel, corrosion-resistance data, fabrication methods, standard sizes and tables of weights and dimensions.

Steel—Lukens Steel Co., Coatesville, Pa. Booklet, 8 pages, 8½x11 in., "Lukens Cromansil Steel," listing physical properties, methods of fabrication and application of an alloy steel having a combination of high strength and good ductility in the rolled condition.

Testing Machines — Tinius Olsen Testing Machine Co., 500 N. 12th St., Philadelphia, Pa. Booklet No. 9, eight pages, describing hydraulic universal testing machines, tensile testing equipment and wire extensometers.

Valve Operators—Philadelphia Gear Works, Philadelphia, Pa. Booklet, 28 pages, 8½x11, "Limitorque Control," illustrating and describing the control system and its method of operation.

BOOKS AND BULLETINS

Applied Hydro- and Aero-Mechanics

O. E. Tietjens, consulting engineer, Westinghouse Electric & Mfg. Co. Based on lectures of L. Prandtl, professor at the University of Goettingen and director of the Kaiser Wilhelm Institute for Flow Research, translated by J. P. Den Hartog, assistant professor of applied mechanics, School of Engineering, Harvard University. 306 pages, 6x9 in. Clothboard covers. Published by the McGraw-Hill Book Co., 330 West 42d St., New York, N. Y. Price $4.

One of the books comprising the "Engineering Society's Monographs" whose publication is sponsored by the four national engineering societies, the American Society of Civil Engineering American Institute of Mining and Metallurgical Engineers, the American Society of Mechanical Engineers, and the American Institute of Electrical Engineers.

This book is a companion to Fundamentals of Hydro- and Aero-Mechanics. The subject matter includes the elements of hydro-dynamics and the laws of mechanical similarity, flow through pipes, boundary layers, drag of bodies moving through fluids, theory of air foil and experimental methods and apparatus for the investigation of hydro- and aero-dynamics problems.

Although it is stated in the preface that the subject matter is presented "in a strictly scientific manner, avoiding a maze of pure mathematical formulas by emphasizing the technical rather than the mathematical treatment." a considerable amount of higher mathematics is involved in the presentation. This book undoubtedly represents the most comprehensive and advanced presentation of this subject in American literature and will be found of great value to engineers and research workers.

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Fundamentals of Hydro- and Aero-Mechanics

O. E. Tietjens, consulting engineer, Westinghouse Electric & Mfg. Co. Based on lectures of L. Prandtl, professor at the University of Goettingen and director of the Kaiser Wilhelm Institute for Flow Research, and translated by L. Rosenhead. 265 pages, 6x9 in. Clothboard covers. Published by the McGraw-Hill Co., 330 West 42d St., New York, N. Y. Price $4.

Representing probably the most advanced book in American literature on this subject, it presents the work of the foremost German scientist and research worker in this field. Quoting from the foreword by Dr. Prandtl, "In order to assure the reader that the portions of the book which are based on my lectures are a faithful presentation of their subject matter, Dr. Tietjens and I agreed that I should look over those parts of the manuscript and, where necessary, bring them into agreement with my ideas as expressed in the lectures. This has been done."

This book contains the theory of the equilibrium of liquids and gases with practical application to balloons and airships and then gives a comparatively vigorous treatment of motion of the "Ideal" liquid with zero viscosity, and ends with a chapter on viscous fluids. The presentation is scientific in all its steps, and of necessity involves a considerable amount of higher mathematics. It is written primarily for the advanced students of the subject who might be engaged in research and development in this field.

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Design of Machine Elements

Virgil Moring Faires. 458 pages, 5½x8½ in. Clothboard covers. Published by The McMillan Co., 60 Fifth Ave., New York, N. Y. Price $4.

Although evidently written for a textbook, the author has departed considerably from the usual form of such books. By numerous references to the practices of recognized companies, the work of research departments, and technical publications, the book partakes of the nature of a reference work.

Relatively a small amount of space is devoted to detailed derivations. Numerous empirical formulae are given, together with charts and tables.

As an example of the comprehensive manner in which the author has treated his subject, Chapter VII on flat belts and pulleys includes all of the general formulae of belt design, together with their derivation, gives tables on the average efficiency of belt joints, the loss of belt capacity in inclined drives, thickness of leather belting, gives a chart on the per cent slip of belts and quotes numerous references to published articles.

On the other hand, practically no information is given on the design of pulleys other than an empirical equation for calculating the face width. The subject matter on pulleys consists of descriptions of various types of pulleys available and the manner in which they should be applied. It should be stated, however, that for screw fastenings, riveting, shafts, gears, bearings, springs, and other machine elements, the design equations are given with considerable detail, in addition to which considerable space is given to commercial considerations and practical requirements in design.

A total of 56 tables and 364 illustrations are included in the book.

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Mechanical Power Transmission

Robert W. Drake. 224 pages, 8½x11 in. 120 illustrations. Paper covers. Published by American Leather Belting Association, 41 Park Row, New York, N. Y. Price 25 cents.

Consisting of a series of 50 reports, these studies by the author cover driven loads, load characteristics of electric motors, fundamentals of belt transmission, and belt efficiency. Tables and charts are given for selecting belt sizes for open drives.

Advantages and disadvantages of pivoted motor bases, V-belts, and silent chain drives are presented in a critical analysis of operating conditions in short-center drives.

Several of the reports describing troublesome applications indicate the operating conditions that the designer must consider in selecting the type and size of pulley and belt for a machine drive. Factors governing the selection of group or individual drive are given for the use of the designer as well as for the plant maintenance man. Specific recommendations are made regarding belt drives for fans, woodworking equipment and compressors.

Presented as the observations of a man well qualified in this field, this series of reports contain much practical design information.

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Thermal Conductivity of Some Irons and Steels

Bureau of Standards, *Research Paper RP-669,* by S. M. Shelton. 10 pages. Typical commercial materials, including cast iron, low-alloy, high-alloy and stainless steel were tested over the temperature range of 100 to 500 deg. C. (212 to 932 deg. F.). At room temperatures the conductivity ranged from 0.665 to 0.148 watts per cm. per deg. C., while at 500 deg. C. the range of coefficients was much narrower, from 0.435 to 0.195. With increasing temperatures the low-alloy steels decreased in conductivity, but high-alloy materials showed an increase. Heat treatment that changed the structure also changed the conductivity. Because of the many and sometimes conflicting factors involved, no general relations were found between conductivity and composition.

Tension and Compression Springs

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IT HAS been found that the ordinary spring formulas for helical compression and extension springs of mean diameter D:

$$P = \pi d^3 s/8D \qquad F = \pi s D^2/G d$$

derived by Reuleaux take no account of an increased fiber stress as the spring diameter is decreased, the size of wire remaining the same. In the formula for actual stress S in terms of load P, outside spring diameter D_o and wire diameter d, the stress s in the Reuleaux formulas is multiplied by a factor y as given by Wahl, which has a practical range from 1.15 to 1.4. That is $S = sy$; and a curve is plotted showing the variation of this factor y with the spring index r_o, which is the ratio of the outside spring diameter D_o to the wire diameter d.

The actual equation of the curve for the y factor is, for practical use, too complex to be embodied in the Reuleaux formulas. But, since only a limited portion of the y factor curve is to be used, it is easy to find and substitute a simple empirical approximate formula that may be so embodied. Such a formula is: $y = r_o/(r_o - 1.42)$ which becomes $y = (r + 1)/(r - 0.42)$ if r is the ratio of the mean spring diameter D to the wire diameter d. Making this substitution, and with $G = 11{,}500{,}000$ for steel springs:

$$P = \frac{0.393\ d^2\ S\ (r - 0.42)}{r\ (r + 1)} \qquad \frac{S\ dr^3\ (r - 0.42)}{3{,}660{,}000\ (r + 1)}$$

It is stated that the best ratio r is equal to 8. It should certainly not be less than 4, and may range to 10. It can be seen that a decrease of this ratio with the same stress and wire size will make a longer spring of smaller diameter, and an increase of same, a shorter spring of larger diameter. Substituting the optimum value of 8 in the two formulas, they reduce to:

$$P = Sd^2/24.2 \qquad F = Sd/68{,}000$$

The chart on the page following presents these equations using two scales at right angles for two of the variables and a family of lines for the third. Each ray from the right-hand focus applies to wire diameters on the left scale, or vice versa, and to corresponding deflections on the upper scale, for one ratio only, there being seven rays for the seven ratios. Each ray from the left-hand focus applies to the same scale of wire diameters on the left and to corresponding loads on the lower scale, for one ratio.

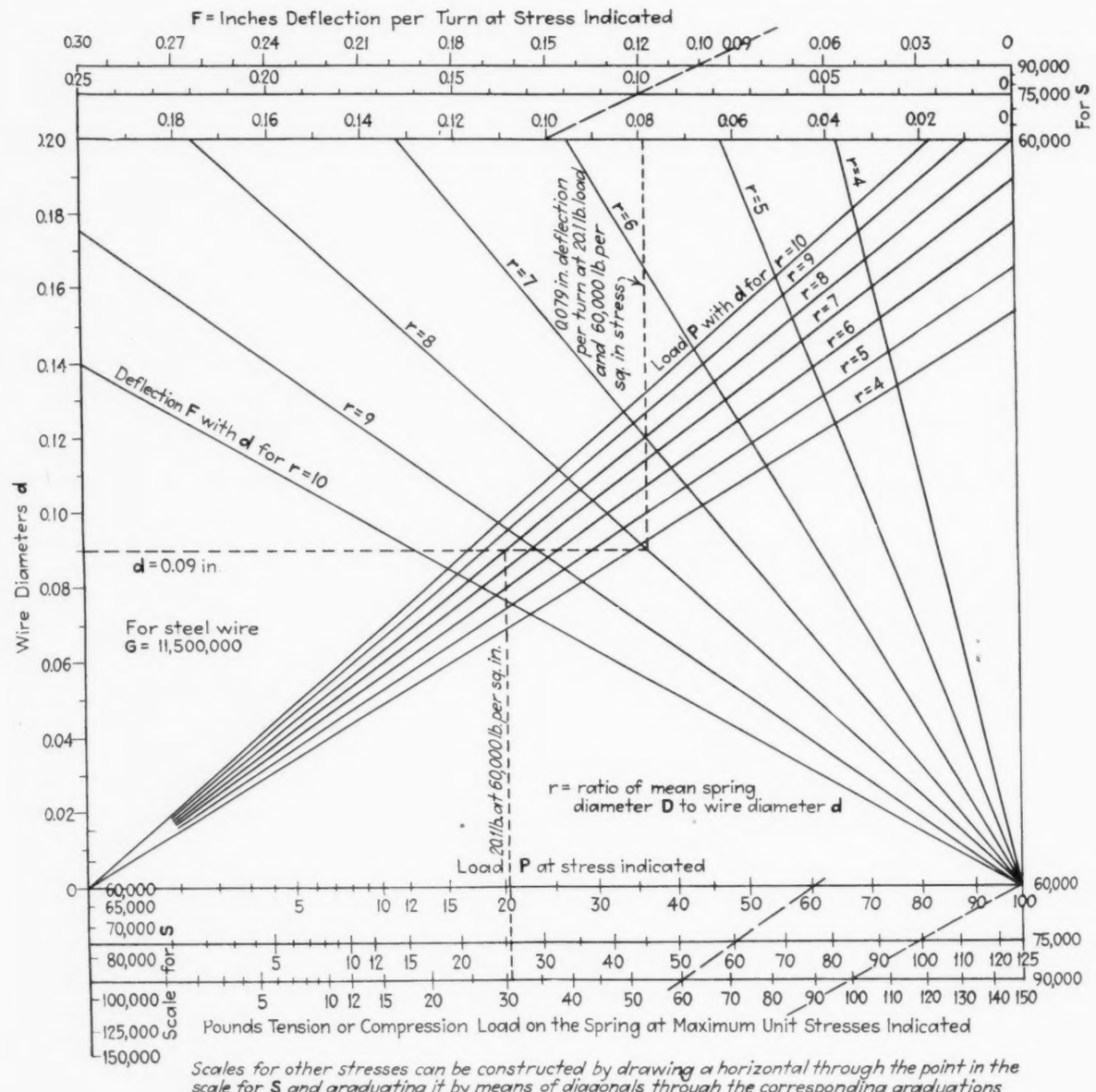
The curves in the chart are for a fiber stress of 60,000 lb. per sq.in. The values of P and F for 75,000 and 90,000 lb. per sq.in. are vertically aligned on the appropriate parallel scale, both above and below the diagram. The location of additional parallel scales for other stresses are shown.

As an example, the index lines show that for a load of 20.1 lb., 0.09 in. diameter wire is required for ratio of 8, from which is calculated 0.72 in. mean spring diameter. The chart also shows that this load will produce a deflection per turn of 0.0795 in. at 60,000 lb. per sq.in. At 75,000 and 90,000 lb. per sq.in. respectively, the load would be 25.1 and 30.2 lb., and the deflections 0.0993 and 0.119 in., as may be seen by prolonging the vertical index line.

As another example, assume that the actual stress S and load P are to be found for a spring, $d = 0.032$ in., $F = 0.092$ in. and $r = 10$. Draw a horizontal at $d = 0.032$ across the diagram. Project its intersection with ray $r = 10$ (from the right-hand

focus) upon F above for scale 60,000, and read $F = 0.046$ in. Now draw a ray through the point $F = 0.092$ on the 60,000 scale to the focus for S scales (upper right) and intersect this with a vertical through $F = 0.046$ on 60,000. Project this intersection horizontally to the right and read on the scale for $S = 120{,}000$ lb. per sq.in. (not shown), which will be the actual unit stress.

Project the intersection of the previously drawn horizontal at $d = 0.032$ with the ray $r = 10$ (from the left-hand focus) upon P at 60,000 below and continue it to intersect the 120,000 S scale drawn horizontally across. Intersect this scale with a ray from the S focus (lower left). Where this cuts the 60,000 P scale at 4.2 lb. is the load for 120,000 lb. per sq.in. stress.



Scales for other stresses can be constructed by drawing a horizontal through the point in the scale for S and graduating it by means of diagonals through the corresponding graduations in the other scales, as shown by the long dashed lines

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